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LUNAR SOIL EXCAVATION SYSTEM

To

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ABSTRACT

Group 4 of ME 4182 undertook the task of designing a lunar soil excavation system. The group decided that two different mechanisms would be necessary to perform the various excavation tasks. One mechanism is used for digging into the surface and the other is used for scraping up layers of the surface. The proposed system will fulfill the expected excavation requirements for NASA's plans.

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1. INTRODUCTION.

Group 4 of M.E. 4182 undertook the task of designing a lunar soil excavation system. The design has to meet the constraints of a harsh lunar environment, limited available power, and high transportation costs to the moon. The group decided that two different mechanisms would be needed to perform the various excavation tasks. One mechanism is used for digging into the surface and the other is used for mining shallow layers of the surface. The moon's gravity is only one-sixth of the earth's gravity. This fact hampers the use of weight dependant traction which is used on the earth. To alleviate this problem both designs used the equal and opposite force principle.

2. EXISTING EQUIPMENT

Several factors involving the lunar environment, reduced gravitational field (weight and traction) and abrasive soil, limit the use of conventional earth-moving equipment for moon excavation. However, several existing design ideas could be incorporated on lunar equipment. In our research of heavy digging equipment; backhoes, clamshells, and tree spades were investigated.

Back hoes and similar equipment offer a variety of soil manipulating capabilities. The hoe is ideally suited for ditch or trench excavation. Utilizing the two piece boom at angles close to ninety degrees, the backhoe operates most efficiently to scoop and move soil or flatten existing piles

with the blade. Calculation of these cycle times takes into account total load, lift, swing, dump, return, and acceleration times for an average figure of 38 seconds(1).

Clamshells are primarily suited for vertical work with loose materials. They are attractive devices primarily due to the direct opposed balance of forces. Utilizing localized opposing forces on a digging implement would help to reduce the dependance on traction for the lunar vehicle. Because clamshells are loose swinging devices, cycle times are higher than for fixed boom equipment due to the distinct acceleration and deceleration phases(1). A tree spade is an attractive device for lunar soil moving applications. The draughting forces associated with the clamshells are reduced, while maintaining locally opposed forces for penetrating blades into the soil. This could possibly reduce the dependance on traction even further.

In research of strip mining equipment, paddle wheel loaders, force feed loaders, augers, and plows were investigated. Paddle wheel loaders consist of front mounted plowshares that funnel loose soil or other materials towards a collecting area where paddle wheels scoop the material to a conveyor. Force feed loaders consist of a blade that is forced through the soil to strip off a layer for collection(1). Conveyors are used to transport the soil to collection bins. In a lunar environment, both the paddle wheel and force feed design ideas could work with the aid of tractive forces.

Since gravitational forces are reduced by a factor of six on the moon, obtaining necessary traction for soil moving activities becomes a problem. For this reason, augers and plows were investigated for their anchoring potential. Augers have the ability to "self-start" into the soil and eventually lock into the surface. Plows self-start into the soil and utilize soil weight and draught to provide resistance. However, both systems would reduce the mobility of the device. The machine would need to stop and re-anchor every time it moved.

The lunar environment and soil characteristics will impose extra demands on excavation equipment. Actuation systems used on digging or mining implements must be able to efficiently and reliably operate under extreme wear conditions such as electrostatic grit, temperature extremes, and vacuum environments. For these reasons, hydraulic and electro-mechanical actuation systems were investigated. Advantages of the hydraulic systems include high mechanical efficiency, quick response times, and accuracy. Some of the disadvantages include a large number of working parts and questionable reliability of system components such as seals and fluids.

Electro-mechanical actuation systems have fewer working parts than do hydraulic systems. Since fluids and seals are not critical system components, reliability of the system is better. However, mechanical efficiency is usually lower for devices such as power screws. Also, response times and accuracy suffer with electro-mechanical actuators due to

inertia effects.

This research into existing types of earthmoving equipment pinpointed several design ideas that were used as a basis for synthesis of the lunar excavation systems, heavy digging and strip mining machines.

3. BACKGROUND OF INTERACTION WITH THE MOON'S SURFACE

Much data about the physical characteristics and mechanical properties of the lunar soil was obtained during the Apollo 14 and 15 missions. Experiments revealing particle size and shape, and soil density, strength, and compressibility provided valuable information for future lunar exploration.

The Apollo 14 voyage provided valuable information about lunar soil through three experiments. Three core samples were taken, three penetrometer tests were performed, and a small trench was dug.

Core samples varied from 7 to 32 cm. in depth. The density of the soil from the samples ranged from 1.45 to 1.60 g/cm³. An analysis of the sample also showed an increasing grain size with the soil depth.

The penetrometer contained a 68 cm. long aluminum shaft .95 cm. in diameter with a 30 degree apex tip. The test consisted of two stages. First, the astronaut pushed the penetrometer with one hand (71 to 134 N.). In the second step, he pushed with two hands (134 to 223 N.). Forces required to insert the penetrometer in the soil are a function

of soil strength, density, land compressibility and of friction between the soil and penetrometer. A general formula for resistance to penetration is

$$F = FS + FP$$

where FS is the skin friction and FP is the point resistance. FS can be considered negligible compared to FP. During the test, the astronaut was able to push the penetrometer an average of 45 cm. with one hand and 86 cm. with two hands.

The trench dug during Apollo 14 was done by Astronaut Shepard. Using a tool similar to a back hoe, he attempted to dig a trench 60 cm. deep with vertical walls. His first cut was about 15 cm. deep with a 70 or 80 degree vertical wall. During later cuts, he was only able to keep the wall at about 60 degrees. Shepard reported the digging to be relatively easy. The final trench was approximately 25 to 36 cm. deep with wall angles of 65 to 80 degrees. Three layers of soil were found during the dig. The color varied with each layer. Grain size increased with depth just as it did with the core tube samples.

The Apollo 15 soil-mechanics experiment was especially informative for several reasons. Not only was the Apollo 15 voyage longer than previous lunar voyages, but the mission used several new experimental devices. The lunar roving vehicle made travel from the lunar module much easier. Analysis of the rover-soil interaction can be applied to the lunar excavation vehicle. The use of a self-recording penetrometer and improved thin wall core tubes provided

valuable information on mechanical and physical properties of lunar soil.

Fully loaded and carrying two crewmen, the rover vehicle weighed 1160 N. on the moon. Considering an average load of 290 N. per wheel, each wheel's unit pressure would be .7 N/cm². The wheels, each powered by an electric motor, had piano-wire mesh tires with a chevron tread covering 50% of the lunar surface contact area. The wheels were 81.5 cm. in diameter and 23.2 cm. in width. The performance of the lunar rover's wire-mesh tires was excellent. The only sources of analysis for the vehicle's performance were crew descriptions, photography, and odometer or speedometer measurements. Even with the vehicle's light weight per wheel (290 N.), the chevron covered wire-mesh wheels had little slippage. The average wheel sinkage into the lunar soil was 1.25 cm. Sinkage increased upwards to 5 cm. on small craters. The astronauts reported no difficulties driving the vehicle on slopes, although the maximum slope traversed was only 12 degrees. The vehicle performed better going up a slope than it did going down a slope. The overall performance of the vehicle was excellent, and a similar mechanism for transporting the excavation vehicle was studied.

The self-recording penetrometer results of Apollo 15 were more accurate than the results of Apollo 14. This accuracy was improved due to the device's ability to measure the forces applied up to 111 N. Results of tests made with a 3.22 cm² base area, 30 degree apex cone showed the average

penetration at 34 N/cm^2 to be approximately 8 cm. An angle of internal friction was estimated to be 49.5 degrees.

The results of these tests by Apollo 14 and Apollo 15 were helpful for two reasons. First, they found accurate values for physical characteristics and mechanical properties of lunar soil. Second, the descriptions of the experiments and the lunar rover's interaction with the soil gave the group a better understanding of a lunar environment, one very different from the environment on earth.

4. SOIL MECHANICS

4.1 Lunar Soil Characteristics. "The lunar soil is generally composed of particles in the silty-fine-sand range and the material possesses a small cohesion and friction angle estimated to be 35 to 40° (3). This description of the lunar soil is one of the more important pieces of information regarding this project. The internal angle of friction is roughly equal to the angle of repose of a soil. This is the slope a loose volume of soil will attain when released, and corresponds to the most extreme volume of digging which would need to be done on the lunar surface.

The density of the lunar soil is fairly consistent, being no greater than 2000 kg/m^3 . The density of soils will increase with depth, but on the moon "the reduced gravity results in a less pronounced increase." The density of the soil did not increase appreciably to a

depth of 80 cm, which makes the computation of soil break-out forces easier (4).

4.2 Shear Stress in Soil. The shear stress characteristics of soil can work both for and against the engineer working in the area of soil engagement. Working with the shear stress properties of soil is similar to working with hydrostatic pressure, as there is an increase in stress with an increase in depth that is in turn directly related to the force necessary to break a volume of soil out of the ground. This force of resistance also affects the traction of a vehicle that is engaging the soil.

Shortcomings of this property include the fact that it takes more force to move a volume of soil than just working against the force of gravity. When digging, compensations must be made for this additional required force.

This shear stress also has advantageous applications. In traction, the greater amount of shear stress the soil can resist, the more traction the vehicle will have. A higher stress is one factor which affects the penetrability of a soil. It is necessary to take this into account when considering the bearing capacity of the soil. The higher the bearing capacity, the more support the soil affords. This is important for the support of machinery against the forces it applies.

The most applicable exploitation of this property to our project is for anchoring purposes. The resistance to

force can be applied to augers which can be buried with less force than they will supply against vertical pulling. Plows or blades can be placed into the soil to resist horizontal forces. In these applications, high shear stresses are advantageous.

A plot of shear stress versus normal stress was developed that had a sufficient range to cover all variations of soil properties as investigated on the moon by the Apollo missions (3). If the shear stress is plotted versus the normal stress, and this point is below the plotted line, there will be sufficient traction for the situation in question (APPENDIX 1). These Mohr's circle-type diagrams for soil failure are similar to those developed by experimentation in terrestrial soil mechanics (5).

4.3 Application Of Soil Characteristics To Mechanical Design.

Predicting the performance of soil cutting blades is complex due to the difficulty in relating stress and strain in soil. Analytical and graphical methods have been used with varying degrees of correlation to experimental data. Therefore, to make the best approximations, cutting blades are made analogous to retaining walls and the theory of passive pressure is applied. The key to accuracy is the assumed shape of the soil failure surface. Then the forces on the blade can be obtained rigorously.

Coulomb developed theories of soil cutting based on ideal dry sand and did not take into account the effects of cohesion and adhesion. He stated that if the shape of the failed soil surface is known, its location can be determined. This statement proved to be very useful in later studies, however, Coulomb's assumed failure shape (a straight line) proved to be inaccurate. Observation of failure surfaces in glass sided boxes revealed a curved profile and theoretical study of ideal materials predicted a surface composed of a logarithmic spiral and a straight line.

Ohde developed a graphical method using the logarithmic spiral and the straight line. The general location of the spiral and line are shown in figure 1. To determine the actual location of the spiral and its intersection with the straight line, a trial and error method must be employed. By choosing several centers for the spiral, the analyst can determine the minimum blade force which represents the actual onset of failure. This graphical method is laborious and time consuming, but the results show good correlation with experimental data.

To enhance the usefulness of the Ohde graphical method, M.S. Osman developed a computer program which would perform iterations choosing different spiral centers. Newton's interpolation formula at unequal intervals is used to determine the minimum value of blade force.

In addition, D. R. P. Hettiaratchi has developed an approximate method for calculating the soil resistance of buried structures. The force required to pull a blade through the soil is given by the general soil resistance equation:

$$P = \gamma z^2 K_\gamma + cz K_{ca} + qz K_q \quad (1)$$

where K_γ , K_{ca} , and K_q are non-dimensional soil resistance coefficients. These coefficients are a function of cohesion, surcharge pressure, soil density, vertical reach of the blade, adhesion, imbedment ratio, and the soil friction angle. For simplicity, the last term in equation (1) can be omitted by neglecting the influence of the uniform surcharge pressure q acting on the soil surface.

The cohesion (c) of the lunar soil is very low (between .05 psi and .20 psi) which aids in the analysis of soil resistance. The second term in equation (1) is due to cohesion and may also be omitted for simplicity.

Mr. Hettiaratchi's analysis assumes a failure surface that can be approximated by a logarithmic spiral with the center at the top edge of the cutting blade. Figure 2 shows the two-dimensional view of the assumed failure surface. The failure area is divided into three zones for the sake of analysis: the "interface zone" (OIJ), the "transition zone" (OJL), and the "modified Rankine zone" (OLMN). We are concerned only with passive

soil resistance pressure because the boundary forces oppose the movement of the cutting blade. Finally, a chart solution was developed based on the retaining wall theory and was used in this project due to simplicity of application. The chart in Figure 3 relates the coefficient of passive pressure (K) to the internal angle of friction and rake angle. The coefficient of passive pressure is used to calculate the passive soil resistance in the following equation:

$$P = \frac{K_r \gamma H^2}{2} \quad (2)$$

This force analysis leads directly to the calculation of the power required for the lunar surface miner.

5. ANALYSIS OF PROPOSALS

5.1 Production Requirements. All presently foreseeable needs for moving lunar material include:

- Digging holes to place living capsules
- Moving soil to bury capsules
- Collection of soil to beneficate for metals, silicates, and oxygen
- Continuous collection for filling sandbags for structural support

As stated in the introduction, two separate vehicles could best fulfill these requirements; one would require little traction for movement of large volumes of soil in a

specified area (diggers), and the other would strip mine continuously as it moved (miner).

An immediate application of a lunar digger would be excavating a hole for a living capsule. To dig a hole to bury half of a 14'x 45' capsule would require the movement of 6,027 cubic feet of soil. To bury it back under a depth of 6 feet of soil would require piling back on top of the capsule 15,396 cubic feet (APPENDIX 2).

The displacement of soil should be done with an automated system that could prepare the site prior to the arrival of astronauts on the lunar surface. The placement and burial of the living capsules could then take place under direct supervision of the astronauts.

The production rate the mining vehicle designers had to keep in mind included the soil needed to beneficiate 1000 metric tons of oxygen per year and to produce the materials to construct buildings on the lunar surface, a figure learned from Barney Roberts at NASA. The amount of soil that needs to be collected to produce the necessary oxygen would be 3285 cubic meters (APPENDIX 3). Other annual soil production requirements would include the collection of soil for use as a structural element in lunar habitats.

5.2 DEVELOPMENT OF DESIGN

5.2.1 Digger. The development of the digging apparatus followed a natural progression of designs based upon

the essential parameter that the operation of digging must not cause the displacement of the vehicle. The one-sixth gravity of the moon affords little normal force to any object on its surface.

The first design idea began with a vehicle having front-to-back or side-to-side scoops, utilizing opposing forces. Difficulties with this design include overcoming a situation where one arm became stuck or could not penetrate the surface of the soil.

A second idea, based on anchoring involved using augers which would be fixed on the undercarriage of the vehicle at opposing 45 degree angles. These screws would dig into the soil, providing a force against which the digging bucket could pull. Set at opposing angles, the horizontal force of the augers would work against each other, rather than against the traction of the vehicle. In addition to problems with soil homogeneity and buried rocks, the resisting force of an augur buried in lunar soil, according to equations suggested by the Civil Engineering department at Georgia Tech, would not be sufficient compared to the power required to bury the blades [APPENDIX 4].

Since anchoring the vehicle limits its range of operation, a variation of the fire-plow proposal became one of the final ideas in this line of

thinking. A vehicle with a bucket arm in front and two fire-plows set at 45 degree angles from the center of the vehicle to the rear would be able to dig. If any of the vehicle's traction were lost, the pulling forward of the vehicle would cause the fire-plows dragging behind to bury themselves. As long as the area of the plows in contact with the soil was greater than the area of the bucket engaging the soil, the resistance to force provided by the plows would be greater than the digging force of the bucket [figure 4]. The vehicle could back-up while lifting the plows to disengage them from the soil.

The other design development originated from the idea of a bucket that would require no opposing forces from the vehicle. Initially a clamshell design was proposed. The clamshell was simple and proven, but it had the disadvantage of requiring a large initial contact area. Thus, it would need a significant downward force to penetrate the blades.

Therefore, we proposed a multi-blade system, similar to a tree spade. As with most tree spades, we used a four-blade design, allowing for small initial contact areas with four points realizing high starting pressures. This allows the bucket to almost self-start with minimal initial downward force.

It soon became apparent that the four-blade design would be fairly complex, requiring a large number of linkage components for actuation. Therefore, a three-blade version was analyzed and found to be sufficient. Originally, a design with flat blades was considered, each driven along a plane into the ground. This configuration still allowed a small initial contact area, and it allowed a simpler actuation technique. The blades were finally curved about the longitudinal axis to provide strength [APPENDIX 5]. The blades will also be individually actuated and monitored to prevent breakage and to allow for the possibility of closing around a rock. This rock lifting capability is an advantage the other proposals did not possess.

It was decided to use two of the three-spade buckets mounted on separate arms. This would allow flexibility in digging arrangements, provide the opportunity to plant one bucket to free the other, and to allow for the continuation of excavation in the event one bucket was damaged.

The blade actuators posed a geometrical problem in that they restricted digging arrangements. They could not clear the soil in certain configurations. Thus, we sought alternative methods of retracting the blades which include rotating the blades upon entry, bringing the blades

to be rolled-up upon retraction, and curving the blades about the transverse axis. It was finally decided to leave the actuators protruding to allow for a more simplified design.

Since hydraulic power is utilized on earth-moving equipment, high flying aircraft, and even the space shuttle, the possibilities of using this type of power for component actuation were investigated.

To manipulate the digging implement, several hydraulic actuators would be employed. Hydraulic cylinders would be used to penetrate blades into the soil. Also, cylinders would be used to control the position of the boom and digging unit. A system of this type would be comprised of motors, pumps, valves, cylinders, lines, and of course the working fluid.

Before sizing these components, many design questions had to be answered. Could a working fluid be found that would withstand the temperature extremes of -140 degrees to +140 degrees Centigrade? Could seals be found that would withstand accelerated wear?

Hydraulic fluids covered under MIL-H-5606 could withstand the hot extreme in temperature but would falter at the cold extreme. Fluids such as chlorotrifluoroethylene have similar characteristics. A means of thermal protection would be required for

system operation at low temperatures.

Although extensive research has been done on suitable sealing materials, no particular type could be found that would withstand the extreme temperatures and wear conditions of the lunar surface(6).

Typical operating pressures for onboard hydraulic systems of high flying aircraft and missiles range from about 425 to 650 kilopascals(7). Using these figures along with the penetrating force requirements of each blade on the digging system, sizing of the cylinders was accomplished.

The horsepower requirements for this system were based on system pressures and typical flow rates for pumps used on tree spade machines (appendix 6).

The figures calculated for cylinder size, horsepower, and kilowatt requirements are misleading. Although the figures are low, they do not take into account problems associated with the reliability of various system components. These problems have the effect of lowering the system efficiency. In addition, fast or extremely accurate operation, usually seen from hydraulics, is not a requirement. For these reasons, hydraulic actuation systems were eliminated as an alternative for power of the proposed digging implements.

After the digging group had narrowed their choices to these two final designs and had understood which actuators could not be used, direct comparison of the two was necessary to narrow the proposal to one system. The decision was based on the factors weight, power, and operating versatility.

Both of the arm/boom arrangements of the machines could be designed to have the necessary reach to excavate living areas on the lunar surface. The arm on the backhoe/fire-plow model had the disadvantage of needing more material to support the force necessary for digging with the tree spade apparatus. The forces working at the end of the tree spade boom include only the weight of the soil, about 500 N, the weight of the power screws and the spade blades themselves, another 1500 N, plus only about 50 N for the digging force itself [APPENDIX 12].

Power needed for the actual breaking into the soil will be about equal for each design. The backhoe/fire plow idea compensates for any lost vehicle traction due to the lunar soil. However, energy is lost in this process.

Versatility of design was also a consideration in the decision between designs. The backhoe/fire-plow design is limited in that it is a fairly

stationary method of excavation. The spade on a boom is translational to many different types of structures, rather than being the rigid, unvarying design of the backhoe/fire-plow.

5.2.2 MINER. One of the functions of the lunar excavation vehicle is strip mining for the purpose of soil collection rather than manipulation. To design a mining system, proposals were made which took into account the lunar environment; special attention was given to the lack of gravity and therefore the lack of conventional traction. Other important considerations were weight due to the high cost of transportation from the earth; maneuverability with a single drive/digging system; and efficient collection without the unnecessary potential energy losses of dropping collected soil. Vehicles were proposed which used opposing motion blades or the addition of weight upon arrival to the moon. Finally, three were looked at in depth and various components were combined for the final proposed mining vehicle.

A vehicle, shown in figure 5, was proposed which used a track with blades to pull the soil towards a slightly submerged fixed blade. The fixed blade provided the dual function of lifting the soil to a collection bin and establishing a resisting force against which the track blades could pull.

The track blades also carried the soil up the length of the fixed blades into the collection bin.

A second idea is shown in figure 6. One cylinder is mounted on each side of the vehicle with the angle between the cylinder and the body of the vehicle variable. Each cylinder would have curved blades spaced such that the soil would be carried to the top of the rotation cycle and dropped into the center of the cylinder. Augers inside the cylinders would carry the soil to the center of the vehicle for collection. Figure 7 shows the side view of one of the cylinders. The angle of the two cylinders could be varied such that they work against each other for digging, and provide a component of force for driving. The cylinders could also be raised in order to use the alternate drive system without digging. One problem with this idea is the relatively large force required to re-position the cylinders. Another problem is the loss of energy when the soil drops from the top of the cylinder to the auger bed. Also, the overall complexity of the vehicle caused us to search for a simpler design.

A third idea (Figure 8) proposed a system which used counter rotating front and rear wheels to dig while providing opposing forces. A forward motion during digging would be obtained because of the 2 to 1 ratio of front to rear digging forces

respectively. The front two front wheels were separated by a distance equal to the width of the single rear wheel which was located centrally so that a complete strip of soil would be mined. The rear wheel could be driven in the forward direction to cause the vehicle to drive instead of dig and could also steer for good maneuverability. A major flaw in this proposal was the problem of efficiently moving the soil from the cutting blades to a collection system. A throwing motion was proposed but the soil trajectory would be hard to control or predict. The idea of steering with the rear wheel was used in the final mining proposal, as was the general concept of a cutting wheel. However, another collection system and method of creating reaction forces was used.

5.3 DESIGN DETAILING FOR THE DIGGER

5.3.1 POWER SCREWS. Power screws will be used to provide linear motion to operate the digging blades and the boom on the lunar excavator instead of hydraulics. Ball bearing screws are similar to power screws except that the friction between the screw and the nut is reduced by ball bearings. They have many advantages over hydraulics:

1. Ball bearing screws are about 90 percent

efficient (8,9). This will reduce the amount of power needed for the digger.

2. Conventional screws will work in temperatures as high as 149 degrees centigrade. This combined with the fact that they are highly efficient and therefor will produce very little heat, means they will not need to be cooled in the lunar environment.

3. The ball bearing screws will not need high pressure seals as hydraulic cylinders do. The only protection from the environment they need is a bellows system to keep dust and dirt off the screw and nut. This could be made of a fabric similar to that found in space suits.

Each screw will be powered by a electric motor attached to the end of the screw. The power used by each motor will be monitored and kept below a preset value to prevent overloading of the beam or blade powered by the screw. Due to thermal contraction of close tolerance parts the screws would not work under the extremely cold conditions found on the moon. A ball bearing screw has been designed to work at temperatures as low as -30 degrees centigrade but, this same screw would not work at

room temperature due to thermal expansion (10). Research will need to be done to determine the best temperature range for a screw to be use on the moon. Resistance heaters could be employed to keep the screws above some minimum temperature. The screws would be lubricated with dry film lube or with MLF-5 (11). On earth they can be used with no lubricant. On the moon, under a vacuum, the metal to metal contact would result in cold welding of the screw to the threaded rod (APPENDICES 7,8).

5.3.2 MATERIALS. The materials used to design the excavation vehicle were an important consideration. Two prime factors were taken into consideration when choosing a material. First, the material should possess the mechanical properties necessary to withstand worst-case forces in a lunar environment. Secondly, the material should be as light as possible to keep earth-to-moon transportation costs down. NASA estimates the cost of transporting material to the moon is \$15000 per pound. Taking these facts into consideration, various materials, most used in the aerospace industry, were studied.

Different materials best suit each part of the excavation mechanism. The arm or boom of the spaded bucket requires high strength characteristics and low density. Several aluminum alloys provide the strengths necessary for the boom. These alloys are

extremely light compared to most steel alloys used in modern back hoe booms. After comparing the various aluminum alloys available, several types were not further considered for use. For instance, Al 7050 possessed good strength characteristics at room temperature, but these strengths were much less at 150°C. One control on the design criteria is that the excavator works at temperatures between -150°C and +150°C. Other aluminum alloys such as 2124 and 2048 are only made in plates; consequently, only mechanical values for plates were available. The decision was narrowed to two alloys, Al 2024 and Al 2215. Both materials perform well at the temperature extremes. Type 2215 actually holds its mechanical properties up to 500°C, but its tensile properties were not as good as type 2024. Because of Al 2024's high tensile properties and ability to withstand the lunar environment, it was chosen as the material for the spade bucket's arm.

Al 2024 is a wrought aluminum alloy containing copper, magnesium, and manganese as hardeners. It is used in many aerospace structures due to its high strength, formability, machinability, and availability. This type alloy has a density of 2.77 g/cm³ (.100 lb/in³). Various heat treatments of type 2024 produce desired mechanical characteristics. The basic temper designations

range from T3 to T8. T8 alloys are best suited for the mechanism's arm. T8 is a solution treated, cold worked, and artificially aged alloy. At room temperature, an Al 2024 tube with T8 temper and .05 to .25 in wall thickness has a tensile yield strength of 56 ksi (APPENDIX 9). At low temperatures, this value is increased, while at high temperatures, the value slightly decreases. However, even at 150°C, the alloy's tensile yield strength is 50 ksi (APPENDIX 10). Properties of Al 2024 are available in the Aerospace Structural Metals Handbook.

The choice of a material for the spade bucket, mining wheels, and mining blade was another consideration. This material will require greater strength and wear resistance due to its application. Weight is also a primary consideration. Titanium alloys provide strength comparable to steel alloys at half the weight. Of the titanium alloys considered, a decision was narrowed to two with desirable characteristics. Both Ti-6Al-4V and Ti-13V-11Cr-3Al have tensile yield strengths of 120 ksi at room temperature and both respond well at the temperature extremes. However, Ti-6Al-4V has higher fracture toughness qualities and was chosen as the appropriate material. The Ti-6Al-4V alloy is the most commonly used titanium alloy and has widespread availability in many forms. Two grades of this

alloy are produced, a standard grade and an ELI (extra low interstitial) grade. The ELI grade has a higher fracture toughness and is often used in cryogenic applications. This fracture toughness is very important due to the spade bucket and mining blade's particular application. Therefore, an ELI grade of Ti-6Al-4V best suits the mechanism. Values of charpy and notch tests show favorable characteristics of the titanium alloy, even at low temperatures. Properties of Ti-6Al-4V can be found in the Aerospace Structural Metals Handbook and the Titanium Handbook.

Annealing a Ti-6Al-4V alloy provides improved toughness although a small loss in strength. Several types of annealing processes are available. Beta annealing provides mechanical characteristics better than the conventional alpha-beta processing. Although beta annealing slightly lowers the yield strength, it improves the fracture toughness, the notched tensile toughness, and the creep strength. The tensile yield strength of an annealed ELI grade alloy is 120 ksi for plate up to one inch thick^(APPENDIX 11). Ti-6Al-4V retains good tensile values over the temperature range experienced on the moon. The tensile yield strength varies from a maximum of 180 ksi at -300° F. to a minimum of 96 ksi at 300° F. These properties are acceptable for

the applications necessary on the excavation vehicle (12,13).

5.3.3 SPADE. The specific design of the spade was dictated by its operation, the material chosen for the blades, the force on the blades, and their controls. It's operation was defined knowing the necessary functions the machine must fulfill.

The placement and operation of the boom and spade will be such that the tips of the blades in the spade will be set into the ground together, so that the axis of the spade is perpendicular to the slope of the soil. This requires that the spade and boom have a great deal of flexibility, justifying the need for 360 of horizontal swivel at the base of the boom, vertical swivel at the first joint, 180 of axial swivel just before the last joint, and one last joint with vertical swivel (figure 9f). The three vertically moving joints in the boom will be actuated with the afore-mentioned power screws, and the axial swivel in the last joint will be accomplished with a small motor mounted on the boom.

The actual design of the spade and it's blades began after the materials were chosen. The force on the blades was calculated to be 625 N [APPENDIX 5], and subsequently the necessary size of the blade will penetrate 48.92 cm. into the soil. Thickness was found to be a maximum of 6.35 cm. [APPENDIX

12,13]. The guide rod size was calculated to be 5.0 cm. in diameter in appendix 14. A cycle time of 48 seconds was developed [APPENDIX 15], with the controls flow diagram in appendix 16.

5.3.4 BOOM. A lattice truss will be the structure providing the movement and support of the spade doing the actual digging. Two beams will guide one spade apiece, each having a capacity of .25 m . The arm/boom design had to encompass the requirements that it has a reach of 10 meters [APPENDIX 2], a support capacity of 2200 N [APPENDIX 8], and be as light as possible. Using the material chosen previously and the calculations in appendix 17, lattice design with hollow tube sections in figure 9 was agreed upon.

The joint allowing flexibility between the boom and the arm will be solid aluminum to provide the maximum amount of strength. The lattice structure will be the support between the joints which will be actuated by the power screws.

5.4 DESIGN DETAILING FOR MINER

5.4.1 Introduction. The final design of the miner involves the best parts of each of the proposed ideas. The miner collects soil by pulling a collecting blade through the soil with a one meter diameter front wheel. The front wheel is 2 meters wide and has 15

centimeter blades that penetrate down into the soil. With the blade on the wheel and the collection blade both down in the soil there are equal and opposite forces set up. The depth of the collection blade is variable so that the most efficient speed is maintained during operation. Power screws are used to adjust the collection blade depth. As soil is forced up the collection blade, it reaches a conveyor which carries the soil to a hopper. The power required to operate the conveyor is dependent on the depth of the collection and the amount of soil it has to carry.

5.4.2 CONVEYOR. The conveyor, shown in figure 10, is the same width as the collection blade and carries soil up into a hopper. A wire and fabric mesh belt is used. The wire is a titanium alloy (TI-) that is very hard so that it will resist wear due to the abrasiveness of the soil and rocks that are scraped up. Rollers under the belt have sealed bearings to keep out dust. Mark's Handbook recommended MoS_2 as a lubricant. A 25 degree slope is used to make the vehicle shorter and more compact. The required power for the conveyor is dependent on the amount of soil to be carried [see appendix 18].

5.4.3 POWER SYSTEMS. The power system consists of two

electric motors. For each wheel there is a separate motor coupled to a planetary spur gear reduction. The motors are to be permanent magnet, series wound, brushless, DC motors. This design was chosen after carefully examining the drive motors that were used on the Lunar Roving Vehicle. The LRV drive motors were of the brush type and it was necessary to seal these motors in a nitrogen environment. The brushless motor that we have chosen was the alternate design for the LRV drive system. This motor was designed by General Electric Co. and was intended to operate in the vacuum of the lunar environment(14,15). The capability of operating in a vacuum simplifies the design by not requiring a sealed cover for the motor. The characteristics sought in the design of the mining vehicle's drive motors are; variable torque, variable speed, light weight, high efficiency, and high torque.

The main drive motor is mounted to the front wheel. This motor is a 5 horsepower brushless DC motor which is coupled to a gear reduction in order to achieve a ground speed of .733 m/sec. There is a secondary drive motor mounted to the rear wheel with the same configuration as that on the front wheel. The rear motor is 3 horsepower and also of the brushless DC type. The horsepower, torque and speed requirements will be examined in appendix 19.

5.4.4 ENGAGEMENT OF COLLECTION BLADE. The collection blade will be positioned at a set 25 degree angle with the horizontal. The blade will be mounted in a set of tracks which will allow the blade to move along the 25 degree line of action yet maintaining adequate rigidity. The depth of the blade in the soil will be infinitely adjustable between 0 and 15 centimeters, the maximum depth of the wheel blades. This depth adjustment will be achieved by the use of two electromechanical linear actuators. These devices consist of a small electric motor which is coupled to a power screw by way of a spur gear reduction. This type of actuator is made by the Warner Brake and Clutch Co.. These devices offer many advantages over the typical power screw arrangement. The actuator has an overload protection device, is completely sealed and will hold a load when power is not supplied to the motor. However, for lunar operation there should be several improvements over the design for actuators used on Earth. The unit needs to be dry lubricated and should be designed to operate in the temperature extremes experienced in the lunar environment. The motors which operate the actuators should also be designed similar to the main drive motors of the vehicle. The actuator and collection blade configuration is shown in figures 11 and 12.

5.4.5 Digging Control. The soil collection rate is a function of both the depth of the force feed blade and the speed of the vehicle. As the vehicle begins to move forward, the force feed blade is lowered some depth into the soil. The wheels now have a resisting force to work against in breaking off soil. The soil is forced up the collection blade, not by conventional traction forces but rather by the front wheel blades themselves. In other words, the force to push soil up the collection blade is transmitted from the wheel blade, through the frame, and to the collection blade, much like the action of a clamshell.

To avoid burying of the front digging wheel, its angular velocity is continuously compared to the velocity of a free rolling, speed indication wheel. As the velocity difference between the two wheels increases, the force feed blade is retracted to increase the absolute speed of the vehicle and decrease digging rate. This trade off between digging and forward progression is continuously modulated by microprocessor to provide efficient operation.

The capability of the rear wheel to drive independent of the front wheel, will provide further control of the digging versus driving ratio. It will allow a greater percentage of front wheel power

to be used for feeding the collection blade and less for pure translation of the vehicle.

5.4.6 Wheels. There are two wheels on our group's proposed lunar soil excavator. The front wheel is two meters wide and the rear wheel is one meter wide. Both wheels are 70 cm. in diameter and are constructed of one cm. thick titanium alloy previously mentioned. The dimensions of the front wheel was arrived at using our soil collection rate estimates (see Appendix 20). The rear wheel is used for steering and was designed more narrow than the front wheel to minimize the required steering torque. The ends of the wheels are sealed with the same titanium alloy and have hubs bolted on each end to interface with the drive system.

5.4.7 Wheel Blades. The blades of our proposed lunar soil excavator are one of the most important components because they provide the traction force necessary to overcome the force required to pull the collector blade through the soil. The length of the blades and the space between each blade are the two critical parameters with which our group was concerned. The dimensions we decided upon were arrived at using both analytical and experimental methods. Mr. Hettiaratch's sub-surface cutting blade analysis, as previously described, provided us with a method to calculate the maximum force

possible on each blade. This force is directly related to the length and width of the blade. Other considerations concerning the length of the blade are the vehicles tendency to roll on top of the soil and the increase in stress at the base of the blade as the length is increased. We decided on a blade length of 15 cm. Experimental methods provided us with a way to decide on the spacing of the blades around the circumference of the wheel.

Our group constructed a small model which we used to help us decide what dimensions to use for the blades. We based our decision for the blade spacing on experimentation with the model in sand and physical intuition. If the blades are to close together, each blade would not have much soil to work against which might cause the wheels to dig down in the soil rather than walk along the surface. If the blades are too far apart, the motion of the vehicle will tend to be more choppy. We decided on twelve blades evenly spaced around the wheel which is a blade spacing of 18.3 cm.. Figure 13a shows the blade length and spacing for one of the wheels.

The stress on each blade was an important parameter in the design of the length and thickness of the blade. The titanium alloy has a tensile yield stress of 6.62×10^8 Pascals at 150 degrees Celsius. The stress calculations for the blades,

found in Appendix 21, are based on the worst case of a blade (15 cm. by 2 m.) that has just entered the soil with a large rake angle. This force acts on the blade $\frac{2}{3}$ of the way down it's length as shown in figure 1, so the blade may be treated as a cantilever beam with a moment arm of 10 cm.. We decided on a blade thickness of 1 cm. which yields a factor of safety of 150. The high factor of safety is to account for impact loads created by hitting rocks. In addition to the high factor of safety, each blade will be reinforced by five ribs, each 1 cm. thick, as shown in figure 14, for additional protection from impact loads.

The base of the blades will be attached to the wheel with 16 long hex head bolts(M14x1.5 by 3 cm.). Nuts will welded to the inside of the wheel to allow for the blades to be replaced quickly and easily serviceable from the outside. The base of the blades must be curved to agree with the curvature of the wheel (see figure 13b). Our group decided each blade requires 16 bolts based on the stress calculations in Appendix 22. These calculations use a worst-case condition in which the blades are not tight against the wheel to provide a factor of safety in shear stress. The factor of safety of 441 associated with the tensile stress accounts for the impact loads involved in hitting rocks. All the

calculations assumed the worst-case condition in which a blade has just entered the soil (see the power calculations in appendix 23). These bolts will be in 8 groups of 2 which are spaced 6 cm. to the left and right of each re-inforcement rib. The bolts are in groups of 2 so the stress involved in rotating the wheels opposite their normal motion will not cause a failure.

5.4.8 Steering Mechanism. Steering is controlled by rotation of the rear wheel about a vertical axis. The main concerns were:

- minimizing bearing loads,
- avoiding interference with the conveyor,
- and drive motor mounting.

These goals were addressed by using a rotating plate mounted in the horizontal plane through the wheel axle (figure 14). The plate is 2 meters in diameter and is a mounting surface for the rear wheel and bearings, and the rear drive motor and reduction unit. The plate rotates on a 2 meter inside diameter roller bearing and is driven by a worm gear meshing with its toothed outside diameter. The large diameter of the bearing surface provides excellent counteracting forces during driving of the rear wheel. The moment arm from the wheel blade to

the axle is approximately .5 meters whereas the moment arm across the supporting bearing is 2 meters. Also, the moment arm for the worm gear is 1 meter which will provide good leverage for steering (see complete CAD drawings in figures 15,16, and 17).

5.4.9 Power Requirements. As mentioned in the soil mechanics section of this report, a chart solution was used to evaluate the forces of soil/blade interaction. Pulling one of the 12 blades on the wheel horizontally through the soil, instead of in a curved path, represents a worst case situation in terms of resistance forces. This scenario was used to provide a factor of safety and because of its direct correlation with the assumed blade motion on the chart solution.

As shown in figure 13a, there are three blades in contact with the soil at any given moment. To account for the forces on the vertical blade as well as the two partially submerged blades, separate equations were used for each. Blades 1 and 2 were analyzed directly by the chart solution (Figure 3). The length of the submerged portion of blades 1 and 3 can be obtained by the following equation:

$$L_{\text{sub}} = L + R(\cos A - 1)$$

where: L_{sub} = Submerged blade length
L = Actual blade length
R = Radius of wheel cylinder
A = Angle between adjacent blades

Blade 3 is working against soil that has already been failed by the blades ahead of it. Therefore, the breakout force determined by the chart method was not used. Instead, the important forces are the weight of the collected soil moving up the stationary blade and the frictional force of that soil. The maximum expected soil volume (on the blade) was calculated by multiplying the surface area of the stationary blade by twice its cutting depth (the cut depth was doubled to account for piling up of soil on the blade). This volume multiplied by the soil density and then by lunar gravity yields the weight of the soil on the stationary blade. The frictional force along the blade is the product of the normal component of the weight and the coefficient of friction for a sand/metal interface. The frictional force was added to the tangential component of the weight to obtain the force on blade 3. The torque on this blade is the product of the total force and the moment arm which is equal to $R + 2L/3$.

Finally the three torque values were combined

to yield the total torque on the front wheel. Having determined a static torque value, we calculated power requirements by multiplying the torque by the angular velocity of the wheel (APPENDIX 23). Again a factor of safety is results by assuming that torque is constant at this maximum value. In actuality, each blade is only sweeping out a portion of the failure surface shown in the figure 13b, because the blade ahead of a given blade has swept away part of soil surface.

The actual amount that an individual blade must remove is dependant on the depth of the fixed resistance/collection blade. If the fixed blade is completely removed from the soil then there will be no resisting force and each blade should enter and leave the soil without significant slippage or cutting of the soil. In this case the vehicle is driving, not digging. As the fixed blade is lowered, more resistance is generated and the blades begin to cut soil as the force to break the soil becomes less than the force to move the vehicle forward. However, all power calculations assume that the vehicle is fixed relative to the ground and thus a maximum force is needed to turn the wheel. All other motions should be accomplished at or below this value.

The calculations for steering power are shown

in appendix 24. To determine the power required to steer, the blade force is calculated as if the blade were being pulled linearly through the soil. Half of this force acts on each half of the blade at $\frac{2}{3}$ the distance from the steering axis. The torque required to turn the wheel is 724 N-m which requires 912 Watts of power at a maximum steering rotation of 12 rpm or 1.26 rad/sec.

6. CONCLUSIONS.

6.1 MINING. The miner has the capability to collect 180 cubic meters per hour assuming no slip of the wheels. However, tests indicated slipping of the wheels which would substantially reduce the amount of soil collected. The mining machine's length to width ratio and low center of gravity make it very stable. The miner could readily lend itself to remote control in which case, stability is essential. Our group built four models which helped determine the feasibility of each proposal and lead to the final design. Some consideration was given to the interchangeability of the wheel blades by using bolt on blades.

6.2 DIGGING. The digger works on a 48 second cycle. Ideally, with both .25 cubic meter buckets in full operation, 30 cubic meters of soil can be moved per hour. This system can be expanded to operate by remote control. Subsequently, preparation of the lunar site will be possible prior to a manned landing.

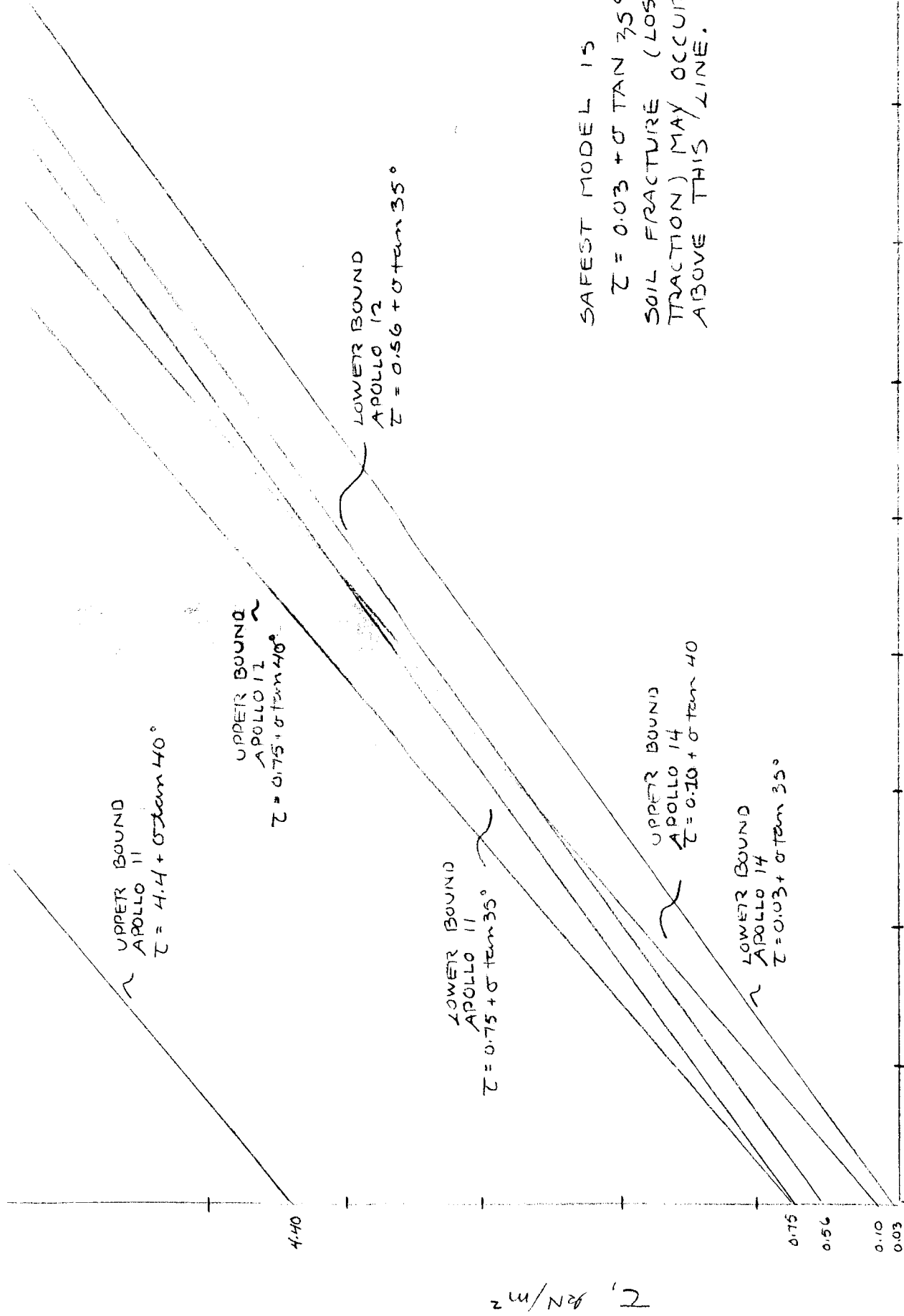
7. RECOMMENDATIONS.

Group 4 recommends that more testing of blade-soil interactions be done to accurately determine the forces on the blades. A more detailed study of how rocks affect the performance of the machines is necessary. The factors of safety are very high for the machines and could be compromised for lighter weight. Each component of the mechanisms should be further analyzed to use the lightest materials possible, including lighter weight electric motors. A separately driven hopper should be considered for the mining system. An ideal vehicle for the lunar surface should be designed to accommodate the digging spade design, as it can be translated to many different vehicle designs.

8. APPENDICES.

- 1) Stress diagram for lunar soil
- 2) Soil movement requirements for burying living capsules
- 3) Soil collection for required oxygen production
- 4) Calculations for normal force to pull out auger
- 5) Calculations for submerging one blade of spade bucket into lunar soil
- 6) Calculation fo hydraulic cylinder size and system horsepower requirements
- 7) Power requirements for ball screws on the spade bucket
- 8) Selection of ball screws for the spade bucket and arm
- 9) Mechanical property limits for extruded tube
- 10) Tensile properties
- 11) Mechanical properties for sheet, strip, and plate
- 12) Three blade spade bucket calculations
- 13) Blade thickness calculations
- 14) Determination of diameter os guide rods on spade bucket actuators
- 15) Cycle time
- 16) Control diagram
- 17) Analysis of boom lattice
- 18) Conveyor power
- 19) Selection of drive motors
- 20) Soil collection rate
- 21) Stress calculation for blades
- 22) Stress calculation for bolts
- 23) Power for wheels
- 24) Power for steering
- 25) Figures

APPENDIX 1 - STRESS DIAGRAM FOR LUNAR SOIL



SAFEST MODEL IS
 $\tau = 0.03 + \sigma \tan 35^\circ$
 SOIL FRACTURE (LOSS OF
 TRACTION) MAY OCCUR
 ABOVE THIS LINE.

$\sigma, \text{ kN/m}^2$

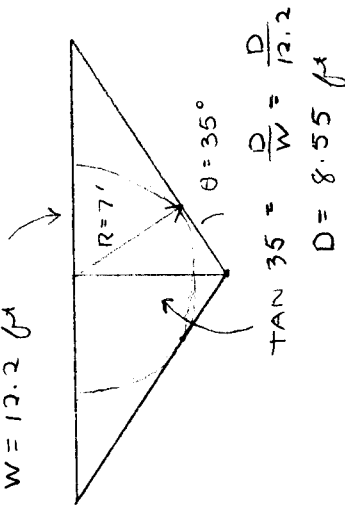
$\tau, \text{ kN/m}^2$

APPENDIX 2

SOIL MOVEMENT REQUIREMENTS FOR BURYING LIVING CAPSULES (ASSUMING THE WORST CASE)

$$\sin 35 = \frac{R}{W} = \frac{7}{W}$$

$$W = 12.2 \text{ ft}$$



$$\tan 35 = \frac{D}{W} = \frac{D}{12.2}$$

$$D = 8.55 \text{ ft}$$

VOLUME OF HALF CONE ON ENDS =

$$\frac{1}{2} \pi \int_0^{8.55} (x \tan 35)^2 dx$$

$$= 666.42 \text{ ft}^3$$

TOTAL VOLUME DUG

$$= 2(666.42) + 4693.95$$

$$= 6026.8 \text{ ft}^3$$

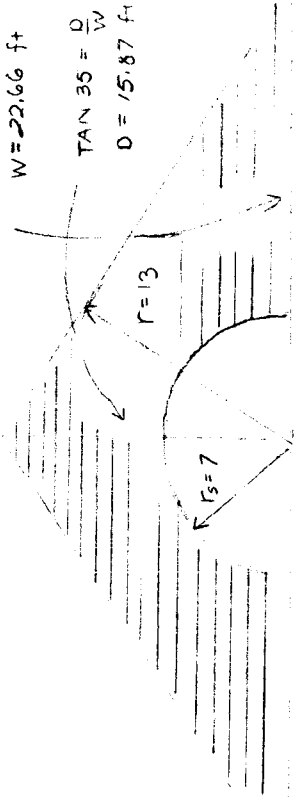
VOLUME REPLACED ABOVE GROUND

$$\sin 35 = \frac{r}{W}$$

$$W = 22.66 \text{ ft}$$

$$\tan 35 = \frac{D}{W}$$

$$D = 15.87 \text{ ft}$$



VOLUME OF HALF CONE ON ENDS

$$= \frac{1}{2} \pi \int_0^{15.87} (x \tan 35)^2 dx$$

$$= 4268.56 \text{ ft}^3$$

VOLUME BETWEEN HALF CONES

$$= 2 \left[\frac{1}{2} (22.66)(15.87)(45) - \frac{\pi}{4} (7)^2 (45) \right]$$

$$= 12,719 \text{ ft}^3$$

VOLUME NOT DISPLACED BY CAPSULE IN TRENCH

$$= 6026.8 - \frac{\pi}{2} (7)^2 (45)$$

$$= 2,563.19$$

TOTAL SOIL MOVED

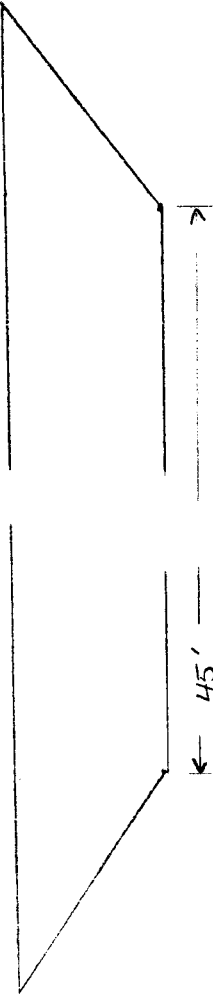
$$= 4268.56 + 12,719$$

$$+ 2,563.19$$

$$= 25,577.6 \text{ ft}^3$$

VOLUME BETWEEN HALF CONES

$$= 2 \left(\frac{1}{2} \right) (12.2)(8.55)(45)$$



APPENDIX 3

SOIL COLLECTION FOR REQUIRED OXYGEN PRODUCTION

The most oxygen-rich ore on the moon is SiO_2 , which makes up (by weight) 41.86 % of the soil at the least, 47.17 % at the most.

Basing oxygen production on the beneficiation of SiO_2 ---

$$\text{SiO}_2 = 44 \text{ g/mole}$$

$$\text{O}_2 = 16 \text{ g/mole}$$

SiO_2 is 36.4 % O_2 by weight.

For every 100 grams of lunar soil,
 $(41.86)(.364) = 15.22 \text{ g}$ is O_2 from SiO_2 .

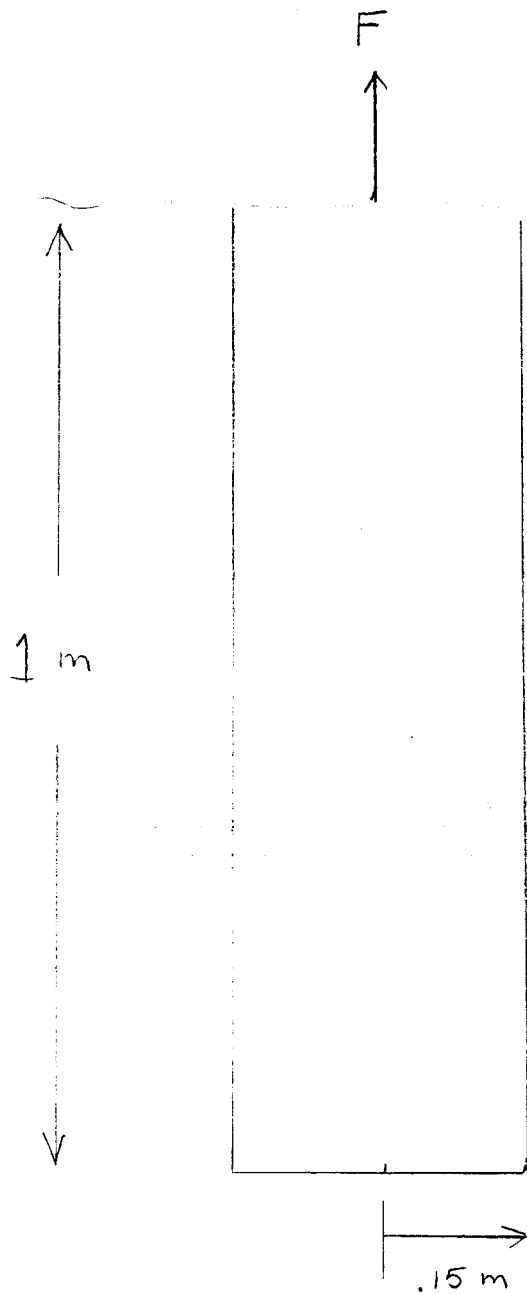
Since 1 m^3 of soil has a mass of 2000 kg, every m^3 of soil contains
 $(2000)(.1522) = 304.44 \text{ kg}$ of O_2 from SiO_2 .

If the oxygen needed is 1000 metric tons per year, annual soil collection to meet this need is ---

$$\begin{aligned} & \left(\frac{1000 \text{ tons}}{\text{year}} \right) \left(\frac{1000 \text{ kg}}{\text{ton}} \right) \left(\frac{\text{m}^3}{304.44 \text{ kg}} \right) \\ & = 3.285 \times 10^3 \text{ m}^3 / \text{year} \end{aligned}$$

APPENDIX 4

CALCULATION OF NORMAL FORCE TO PULL OUT AUGER



$$Z = 1 \text{ m}$$

Vertical stress at a depth

$$\begin{aligned} \gamma &= \sigma_v \\ &= \rho g z \end{aligned}$$

$$\begin{aligned} \text{Horizontal stress at that depth} &= \sigma_H \\ &= k \sigma_v \end{aligned}$$

$$k = 0.5 \text{ for sandy soil}$$

Break-out force will be the integral of the stress over the area in contact

$$= F$$

$$= \int \sigma_H P dz$$

$$= \int_0^1 k \rho g z (2\pi)(.15) dz$$

$$= \int_0^1 (0.5)(2000 \text{ kg/m}^3)(1.63 \text{ m/sec}^2)(2\pi)(.15) z dz$$

$$= 769.69 [z^2]_0^1$$

$$= 769.69 \text{ N}$$

APPENDIX 5

Calculation For Submerging One Blade of Spade Bucket into Lunar Soil

Assumptions:

- 1) Draft angle = 35°
- 2) Extreme case of blade almost completely submerged.
- 3) Only normal force on blade resists motion

Given:

- 1) Volume of entire 3-spade bucket, $V_T = 0.25 \text{ m}^3$
- 2) Density of soil, $\rho = 2000 \text{ kg/m}^3$

- Weight of soil in total bucket: $W_T = V_T \rho g = (0.25)(2000)\left(\frac{9.8}{10}\right)$
 $W_T = 816.67 \text{ N}$
- Weight of soil above each blade: $W_B = W_T/3 = 272.22 \text{ N/blade}$
- Normal force on one blade from soil: $N_B = W_B \cos 35^\circ = 222.99 \text{ N/blade}$
- Coefficient of Friction on blade: $\mu = \frac{2}{3} \tan 35^\circ = 0.467$
- Frictional force on one side of blade: $F_f = \mu N_B = 104.14 \text{ N/side}$
- Total frictional force on blade: $F_T = 2F_f = 208.27 \text{ N}$
- Thus, force required to submerge blade is at most F_T . But, let us introduce a safety factor to account for density variations and the weight of the bucket effecting the underside normal force. Let $n=3$.
- So, $F = nF_T = 3(208.27 \text{ N}) = \underline{624.8 \text{ N}}$

Appendix 6 - calculation of hydraulic cylinder size and system horsepower requirements.

- ① System operating pressure @ pumps - 3000 - 4000 psi (650 kPa - 850 kPa)
- Estimation of system losses 500 psi (~100,000 Pa)
- ② Force required for each blade ~ 200N x 1.50 = 325N

③ Area of cylinder:

$$A = F/P$$

$F = \text{force}$
 $P = \text{pressure}$

$$A = \frac{325 \text{ N}}{550,000 \text{ Pa (N/m}^2\text{)}} = .00061 \text{ m}^2$$

④ Determine cylinder bore:

$$\frac{\pi d^2}{4} = .00061 \text{ m}^2$$

$$d \approx 27.8 \text{ mm}$$

- horsepower requirements (system) -

- ① Typical flow rates from pumps used on tree spade devices:

$$Q = 5-12 \text{ gpm } (.00032 \text{ m}^3/\text{s} - .00075 \text{ m}^3/\text{s})$$

- ② Pressures at pumps - 650 kPa

- ③ Governing equation for fluid HP -

$$\text{fluid HP} = \frac{PQ}{75}$$

$P = \text{pressure}$
 $Q = \text{flow rate}$

$$= \frac{(650,000 \text{ N/m}^2)(.00075 \text{ m}^3/\text{s})}{75}$$

$$= 6.50 \text{ HP}$$

- ④ Determination of kilowatts of power input:

$$\text{BHP} = \frac{\text{FHP}}{\eta}$$

FHP - fluid horsepower
BHP - brake horsepower
 η - efficiency

Appendix G cont.

$$BHP = \frac{6.50 \text{ HP}}{.75}$$

η typical .75, for pump-motor tandem.

$$= 8.67 \text{ HP}$$

1 kilowatt = 1.341 horsepower

$$\frac{8.67 \text{ HP}}{1.341 \text{ HP}} = 6.47 \text{ kW}$$

Appendix 7

Power Used by Ball Screws to Insert the Blades

Assumptions:

1. The soil cutting blades will take ten seconds to move into the soil.
2. The power screws are 90% efficient [Ref. 8,9].

$$\text{Force} = \frac{209.6 \text{ N}(3)}{4.448 \text{ N/lb}} = 141.3 \text{ lb}$$

$$\text{Power} = \frac{141.3 \text{ lb} (33.5 \text{ inch})}{\left(\frac{550 \text{ ft}\cdot\text{lb}}{\text{sec}\cdot\text{hp}} (10 \text{ sec}) \right) \frac{\text{lb}}{12 \text{ in}}} = .0717 \text{ hp} \\ = 60 \text{ Watts}$$

$$\text{Total Power} = 3 \text{ blades } (60 \text{ W}) = 180 \text{ W}$$

Appendix 8

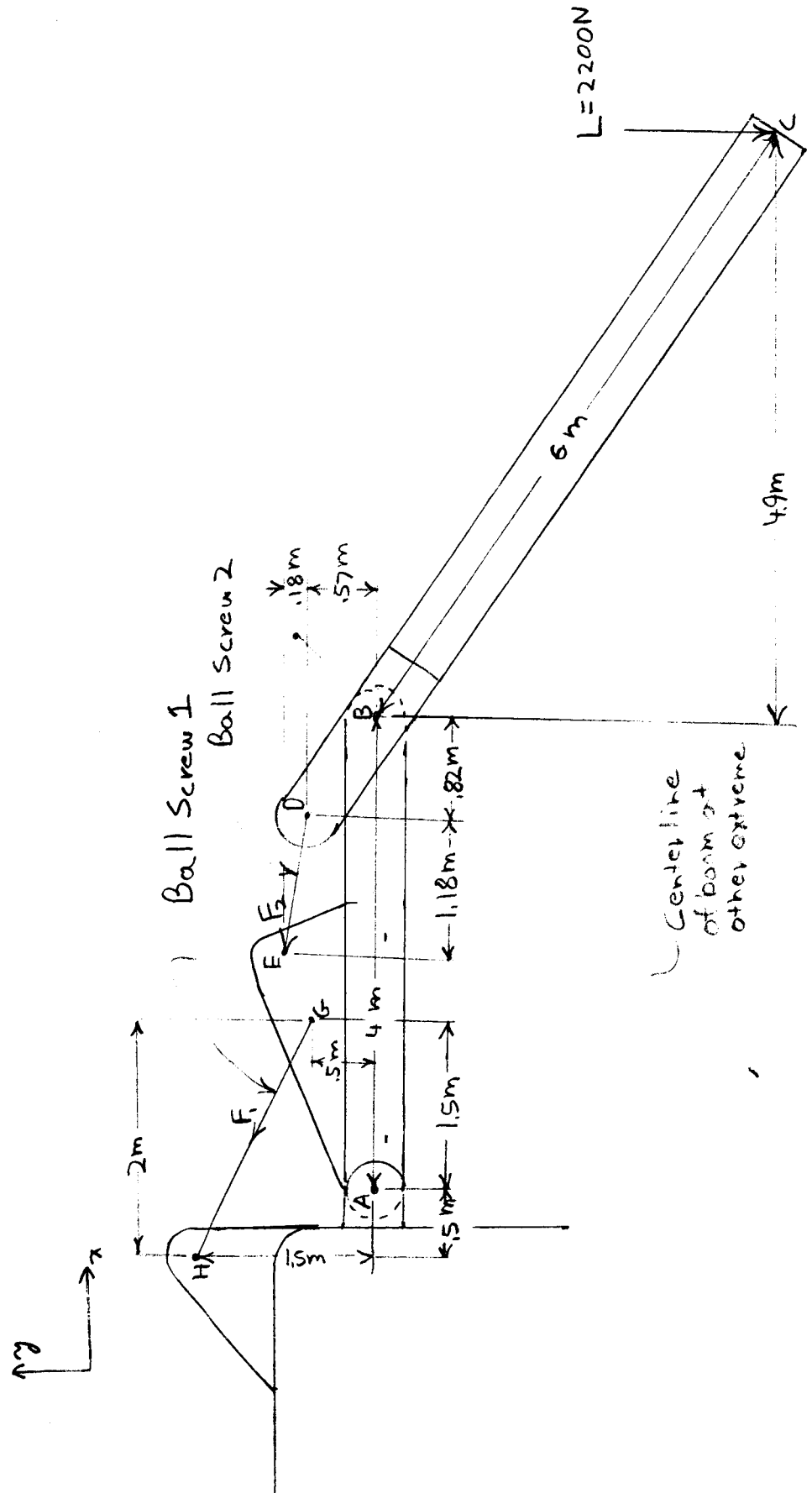
Selection of Ball Screws

Assumptions:

1. The maximum weight of the soil collector plus the soil is 2200N on the moon.
2. The lubricant used will give the ball screws the same life expectancy as found on earth.
3. The travel rate for screws moving the arm will be below 153cm/min.
4. The soil cutting blades will take ten seconds to move into the soil.
5. All the screws will be designed for one million cycles along their entire length at their maximum possible load.

Appendix 8

Sketch of beam in the position that will put the maximum load on the ball screws



Appendix 8

Selection of Ball Screw 1:

Find force F_1 :

$$\sum M_A = 0 = 2200N(4+4.9) - F_1 \left(\frac{2m}{\sqrt{(2m)^2 + (1m)^2}} \right) (1.5m) \\ - F_1 \left(\frac{1m}{\sqrt{(2m)^2 + (1m)^2}} \right) (1.5m)$$

$$F_1 = 17512N = 3939lb$$

Find the maximum inches of travel by the nut in one million cycles of the arm!

$$\text{travel per cycle} = 2(\sqrt{2m^2 + 1m^2} - 1.5m) = 3.5m \\ = 135in,$$

$$\text{travel in 1 million cycles} = 135 \times 10^6 in$$

Using the "Life Expectancy for Non-preloaded Rolled Thread Screws" table on page 9 of section C in the 1979 Linear Industries Ltd. Catalog, it is found that a screw with a 2.25 in (5.7cm) and .5 in (1.28cm) lead will work in this application.

The screw will always be in tension so it will not buckle.

Appendix 8

Selection of Ball Screw 2

Find force F_2 :

$$\begin{aligned}\sum M_B = 0 = & 2200\text{N}(4.9\text{m}) + F_2 \left(\frac{.18}{\sqrt{1.18^2 + (.18)^2}} \right) .82 \\ & - F_2 \left(\frac{1.18}{\sqrt{1.18^2 + .18^2}} \right) .57\end{aligned}$$

$$F_2 = 24515\text{N} = 5511.1\text{lb}$$

$$\text{travel per cycle} = 2(1.5\text{m}) = 3\text{m} = 117\text{in}$$

$$\text{travel in one million cycles} = 117 \times 10^6 \text{in}$$

Using the "Life Expectancy for Non-preload Rolled Thread Screws" table, it is found that a screw with a diameter of 2.25in (5.7cm) and a lead of 1in (2.56cm) will work for this application.

Appendix 8

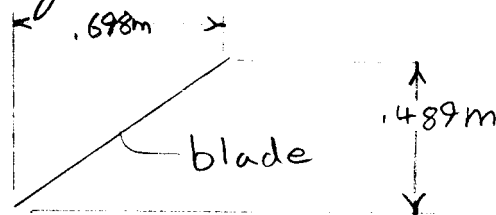
Selection of the Ball Screws used to power the blades:

The calculated force = 209 N

The factor of safety is 3.

$$\text{Design force} = 209 \text{ N} (3) = 627 \text{ N} \\ = 140.9 \text{ lb}$$

The length of travel:



$$\text{length of travel} = \sqrt{(.698\text{m})^2 + (.489\text{m})^2} = .852 \text{ m} \\ = 33 \text{ in}$$

$$\text{travel per cycle} = 66 \text{ in}$$

$$\text{travel per } 10^6 \text{ cycles} = 66 \times 10^6 \text{ in}$$

Ball screw size based on life expectancy from life expectancy table is:

$$\frac{1}{2} \text{ in dia} = 1.27 \text{ cm dia}$$

$$\frac{1}{2} \text{ in lead} = 1.27 \text{ cm dia}$$

$$\text{time to put blade in ground} = 10 \text{ sec}$$

$$\text{travel rate of nut} = \frac{33 \text{ in stroke}}{10 \text{ sec}} \frac{60 \text{ sec}}{1 \text{ min}} = 198 \text{ in/min}$$

Appendix 8

The screws will be supported on both ends and will have a maximum length of 36 in between supports. From the table, "Travel Rate vs Length for Designated Ball Screws", on page 8 of section C in the 1978 Linear Industries Ltd. Catalog it is found that the screw found above will be large enough.

The screws pushing the blades in will be under compression and therefore must be large enough to prevent buckling. From the table on page 7 of section C of the catalog mentioned above the screw found above will be large enough.

Alloy	2024				
Form	Drawn Table				
Condition	Wall Thickness, inches	Area, in. ²	F _{ty} , ksi (Min)	F _{tu} , ksi (Min)	Elongation, e, (2 inch or 4D), percent, (Min)
T81, T8510 & T8511	0.050-0.249	All	56.0	64.0	4
	0.250-1.499	All	58.0	66.0	5
	1.500 and Over	Up Thru 32	58.0	66.0	5

TABLE 3.013. ALUMINUM ASSOCIATION MECHANICAL-PROPERTY LIMITS FOR EXTRUDED TUBE (16)

AEROSPACE STRUCTURAL METALS HANDBOOK

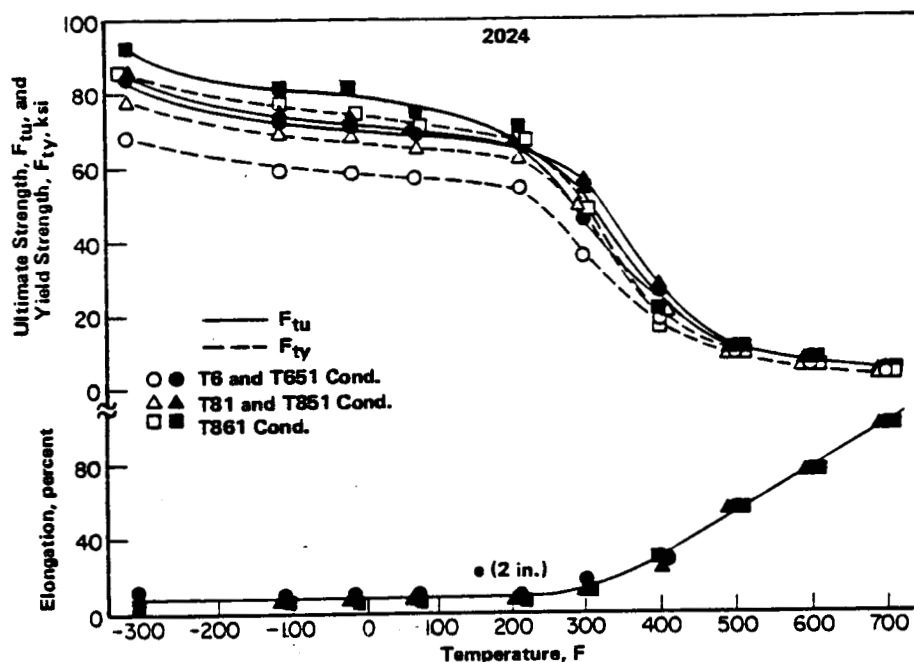


FIGURE 3.0315. TYPICAL TENSILE PROPERTIES AT TEMPERATURES FROM -320 TO 700 F FOR WROUGHT PRODUCTS IN SEVERAL HEAT-TREATED CONDITIONS AFTER 10,000 HOURS' EXPOSURE TIME AT TEMPERATURE (16) (26)

AEROSPACE STRUCTURAL METALS HANDBOOK

Alloy	Ti-6Al-4V								
Form	Sheet, Strip, and Plate (Normal Purity)					Sheet, Strip, and Plate(a) (ELI Grade)			
Condition	Annealed					Annealed			
Diameter or Thickness, in.	< 0.008	0.008 to 0.025	0.025 to 0.063	0.063 to 0.1875	0.1875 to 4.000	0.008 to 0.015	0.015 to 0.025	0.025 to 1.000	1.000 to 3.000
F _{tu} , ksi (Min)	134	134	134	134	130	130	130	130	130
F _{ty} , ksi (Min)	126	126	126	126	120	120	120	120	115
e (2 in. or 4D), percent (Min)	—	6	8	10	10	6	8	10	10

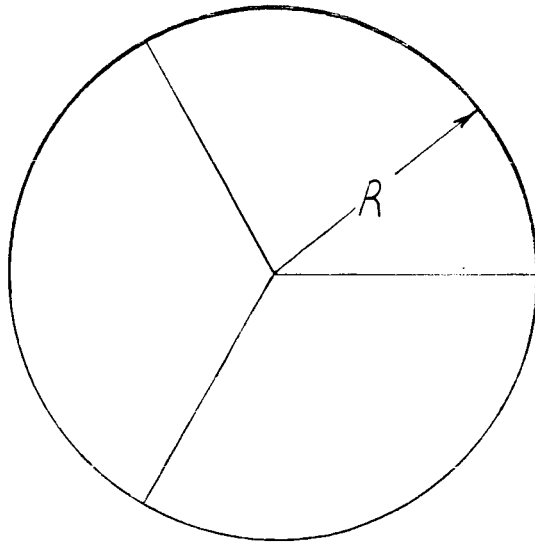
(a) Using specified tensile test specimens, product tested at -423 F shall maintain a notch/unnotched tensile ratio of not less than 0.75 (A2).

TABLE A3.0111. AMS SPECIFIED MECHANICAL PROPERTIES FOR SHEET, STRIP, AND PLATE (A2, A3)

AEROSPACE STRUCTURAL METALS HANDBOOK

APPENDIX 12

Three Blade Spade Bucket Calculations



$$\frac{R}{h} = \tan 55^\circ \Rightarrow R = h \tan 55^\circ$$

$$V = \frac{1}{3} = A_B h = 0.25 \text{ m}^3$$

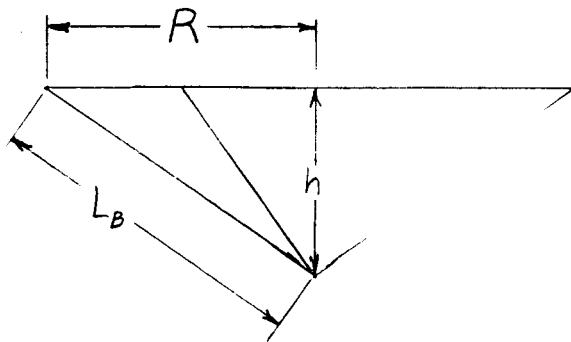
$$\text{so, } 0.75 = \pi R^2 h$$

$$\text{also, } 0.75 = \pi h^2 \tan^2 55^\circ h$$

$$\Rightarrow h = 0.4892 \text{ m.}$$

$$\therefore R = 0.6986 \text{ m}$$

$$\text{and, } L_B = \frac{h}{\cos 55^\circ} = 0.8529 \text{ m.}$$



The weight of the bucket is found by:
 $W_B = \text{dens.} \cdot \text{gravity} \cdot \text{Volume} = d \cdot g \cdot \frac{1}{3} \cdot (R_o^2 - R_i^2) \pi h$

The maximum thickness of the blade is, $t_a = 6.35 \text{ cm}$
 then $R_i = R - t_a = 0.6986 - 0.0635 = 0.6351 \text{ m}$

So, the approximate weight of the three blades is:
 $W_B = 4,470 \frac{\text{N}}{\text{m}^3} \cdot \frac{9.8 \text{ m}}{9.8} \cdot \frac{1}{3} \cdot (0.6986^2 - 0.6351^2)$

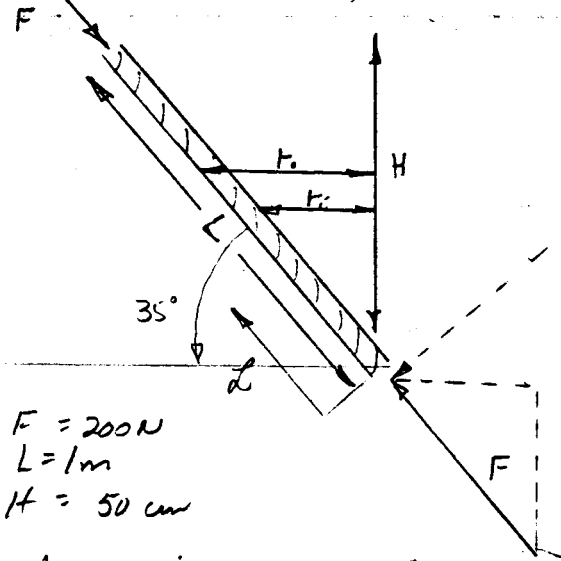
$$\cdot \pi \cdot 0.4892 = 316.76 \text{ N}$$

$$U = \frac{1}{3} h (B_1 + B_2 + \sqrt{B_1 B_2})$$

Appendix 13 -

Blade thickness calculations

Side section of cone blade



$$P = 3F = 600 \text{ N}$$

torsional component of horizontal force increased by a factor of 3 to account for obstructions in the soil.

$$F = 200 \text{ N}$$

$$L = 1 \text{ m}$$

$$H = 50 \text{ cm}$$

Governing equation:

$$(1) \sigma = \frac{Mc}{I} \quad (a) \quad M = P \times l \quad \text{where } l = \left(\frac{r_o - r_i}{2} \right) \left(\frac{1}{\sin 55^\circ} \right)$$

$$(b) \quad I = \frac{\pi}{192} (16r_o^4 - 16r_i^4)$$

$$(c) \quad c = \left(\frac{r_o - r_i}{2} \right)$$

$$(2) \therefore S_y \geq \frac{600 \text{ N} \left(\frac{r_o - r_i}{2} \frac{1}{\sin 55^\circ} \right) \left(\frac{r_o - r_i}{2} \right)}{\frac{\pi}{192} (16r_o^4 - 16r_i^4)}$$

$$\text{where } S_y = 129,000 \text{ psi} = 827,376,000 \text{ Pa}$$

$$\therefore 827,376,000 \text{ N/m}^2 \geq \frac{699.45 (r_o - r_i)^2}{(r_o^4 - r_i^4)}$$

Appendix 13 - cont.

Let $R = .0625 m$

$R_o = .0512 m$

$$8.13 - 1,182,395 R_i^4 \geq R_o^2 - 2R_o R_i + R_i^2$$

$$8.1274 \geq 1,182,395 R_i^4 - .1024 R_i + R_i^2$$

Try values for R_i :

* $.04485 m$

$$8.1274 \geq 4.782$$

factor of 1.70 (.25" thickness)

$.04803 m$

factor of 1.29

Let $R = .25 m$

$R_o = .2048 m$

$$2,080.97 - 1,182,395 R_i^4 \geq R_o^2 - 2R_o R_i + R_i^2$$

$$2,080.93 \geq 1,182,395 R_i^4 - .4096 R_i + R_i^2$$

Try values for R_i :

$.19845 m$

factor of 1.13

Try smaller

$.19528 m$

factor of 1.21

* $.18575 m$

factor of 1.47
(.75" thickness)

Let $R = .75 m$

$R_o = .6144 m$

$$168,447.5 \geq 1,182,395 R_i^4 - 1.288 R_i + R_i^2$$

Try values for R_i :

$.59535 m$

factor of 1.13

$.57630 m$

factor of 1.29

Appendix 13 - cont.

values for r_i
* .5636 m

factor of 1.42 (2" thickness)

Let $L =$

$r_b = .6990 m$

$$282,273.78 \geq 1,182,395 r_i^4 - 1,398 r_i + r_i^2$$

Try values for r_i

.6355 m

factor of 1.46 (2.5" thickness)

Summary of values

L	thickness
.0625 m	.25" (.00635 m)
.25 m	.75" (.01905 m)
.75 m	2.0" (.05080 m)
?	2.5" (.06350 m)

APPENDIX 14

Determination of Diameter of Guide Rods on Spade Bucket Actuators

Assumptions:

1) Ti-6Al-4V construction, solid round x-section,
and $S_{ey} = 120 \text{ ksi}$ @ room temp. (at 300°F , $S_{ey}' = S_{ey} \cdot 0.80$)

2) Maximum normal force at blade tip, $P = 650 \text{ N}$.

3) Bending stress causes failure.

Given:

1) Length of blade, $L_B = 0.8528 \text{ meters}$.

2) Length of guide rods, $L_R = 1.100 \text{ meters}$.

• Moment of inertia, $I = \frac{\pi d^4}{64}$

• Bending stress, $\sigma = \frac{Mc}{I} = \frac{64Mc}{\pi d^4}$ ($c = \frac{d}{2}$)

• Max. moment on rods (2 per blade)
is when guide block is halfway up
guide rods.

So, $M_{\text{blade}} = L_B \cdot P = 1.10 \cdot 650 = 715.0 \text{ N}\cdot\text{m}$

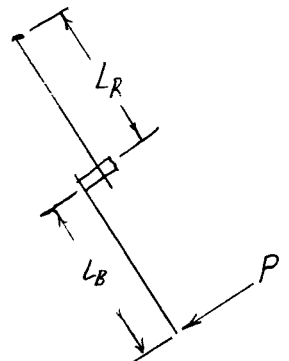
$M_{\text{rods-}\frac{1}{2}} = \frac{M_{\text{blade}} \cdot x}{L_R}$ (max. when $x = L_R/2$), $M_{\text{rods}} = \frac{715.0}{2} = 357.5 \text{ N}\cdot\text{m}$

• Then, moment on each rod: $M_R = \frac{M_{\text{rods}}}{2} = \frac{357.5}{2} = 178.75 \text{ N}\cdot\text{m}$

• $S_{ey}' = 0.80 (120 \times 10^3) = 6890 = 661.4 \times 10^6 \text{ Pa}$

$\therefore 661.4 \times 10^6 = \frac{64(178.75) \frac{d}{2}}{\pi d^4} \Rightarrow d = 0.140 \text{ m} = 1.40 \text{ cm}$

• Let $n = 3.5$, so, $d' = nd = 3.5(0.140) = 0.0491 \approx \underline{5.0 \text{ cm}}$



CYCLE TIME

DIGGING APPARATUS

LOAD TIME 10s

LIFT & RETRACT
TIME 5s

SWING TIME 4s

EXTEND & DROP
TIME 5s

DISCHARGE TIME 10s

LIFT & RETRACT
TIME 5s

SWING TIME 4s

EXTEND & DROP
TIME 5s

TOTAL
48s

Appendix 15 - cont.

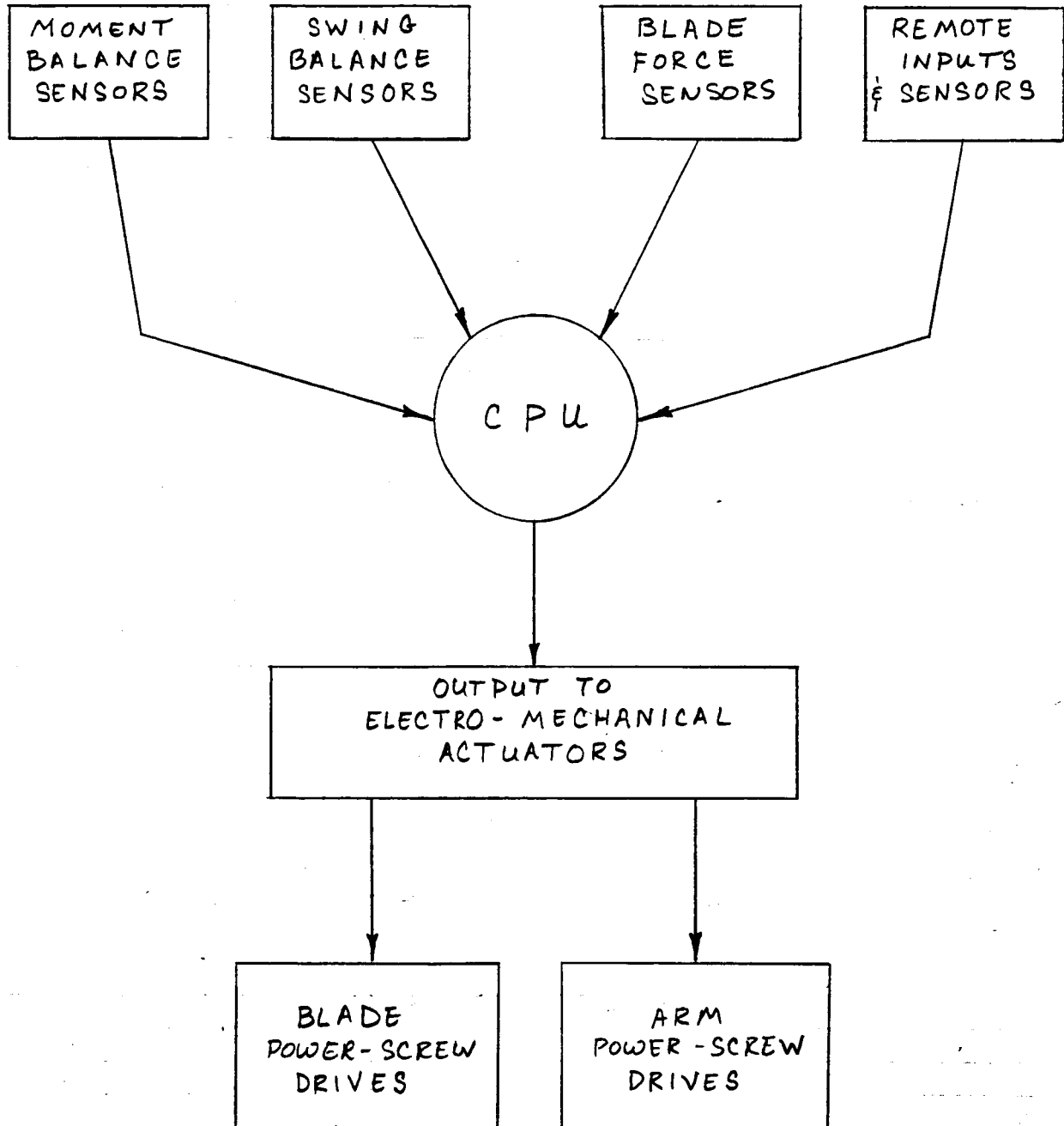
Swing time estimates:

$$\left. \begin{array}{l} \text{Assume } 5 \text{ rpm} = .52 \text{ rad/s} \\ \text{ \& } 120^\circ \text{ swing angle} \end{array} \right\} t = \frac{2.09 \text{ rad}}{.52 \text{ rad/s}} = 4.0$$

Times are estimated using typical estimates for similar earth-moving equipment.

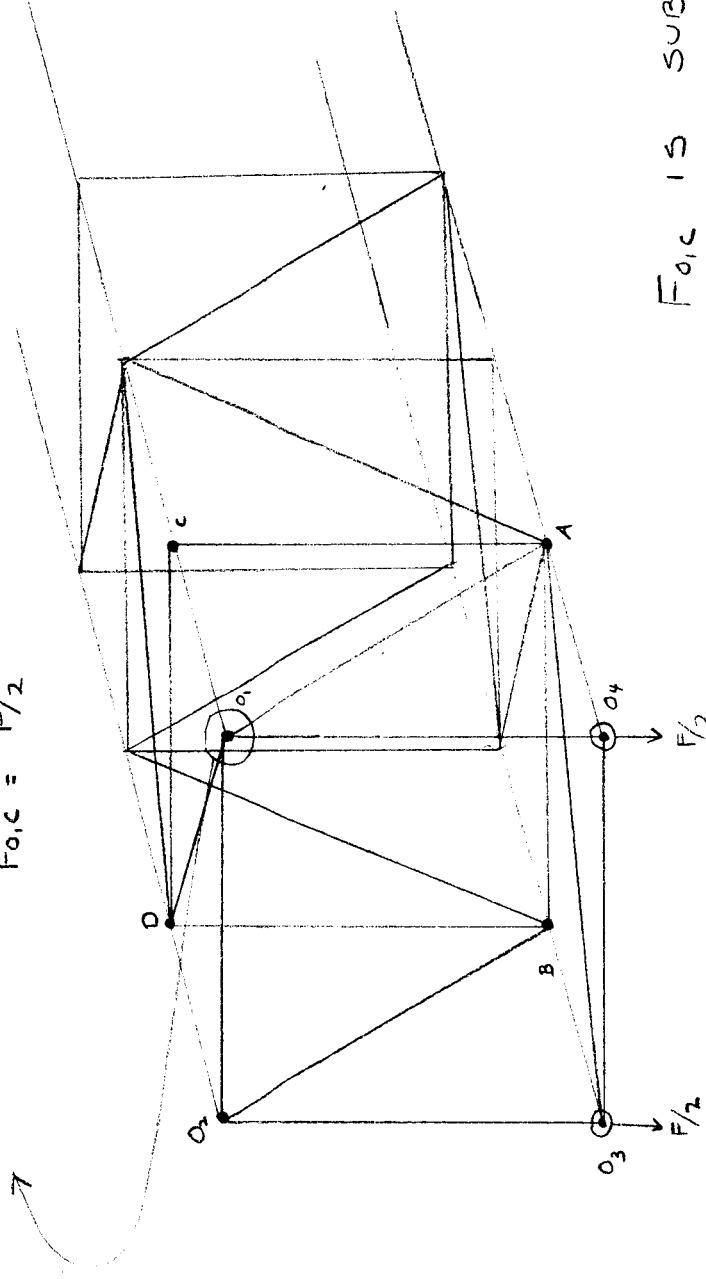
CONTROL DIAGRAM

DIGGING APPARATUS



APPENDIX 17- ANALYSIS OF BOOM LATTICE

$$\begin{aligned}\sum F_x &= -F_{0,02} = 0 \\ \sum F_y &= -F_{0,1c} + F_{0,1A} \cos 45^\circ = 0 \\ \sum F_z &= -F/2 + F_{0,1A} \sin 45^\circ = 0 \\ F_{0,1A} &= .7071 F \\ F_{0,1c} &= F/2\end{aligned}$$



$F_{0,1c}$ IS SUBJECT TO STRESS DUE TO BENDING, SO DESIGN A LATTICE MEMBER THAT WILL WITHSTAND A FORCE OF $F/2$. F IS THE NORMAL FORCE OF THE SPACE.

APPENDIX 17 (CONTINUED)

ALUMINUM ALLOY PROPERTIES

$$E = 10.5 \times 10^3 \text{ ksi}$$

$$= 72.345 \text{ GPa}$$

$$\rho = 2770 \text{ kg/m}^3$$

$$S_y = 50 \text{ ksi}$$

$$= 344.5 \text{ MPa}$$

SOLID MEMBERS IN LATTICE

$$S_y = \eta \sigma_x = \eta \frac{(F/2)(32)l}{\pi d^3}$$

$$\text{ASSUME } F = 2.5 \text{ kN}$$

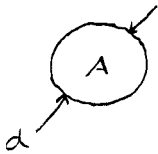
$$\eta = 3$$

$$l = 500 \text{ mm}$$

$$.3445 = \frac{3(1.25)(32)(500)}{\pi d^3}$$

$$d = 38.13 \text{ mm}$$

$$A = \pi/4 d^2 = 1141.9 \text{ mm}^2$$



HOLLOW MEMBERS IN LATTICE

$$S_y = \eta \sigma_x = \eta \frac{F/2(32)ld}{\pi(d^4 - d_i^4)}$$

$$\text{ASSUME } F = 2.5 \text{ kN}$$

$$l = 500 \text{ mm}$$

$$\text{IF } t = 5 \text{ mm and } d = 45 \text{ mm}$$

$$d_i = d - 2t = 35 \text{ mm}$$

$$\sigma_x = \frac{(1.25)(32)(500)(45)}{\pi(45^4 - 35^4)} = .11019$$

$$\eta = S_y / \sigma_x = 3.12$$

$$A = \pi/4 (d^2 - d_i^2) = 628.32 \text{ mm}^2$$

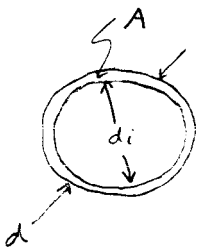
THIS IS MUCH LESS THAN THE AREA OF A SOLID MEMBER, WITH AN EVEN GREATER SAFETY FACTOR.

THE AREA CAN BE REDUCED FURTHER --

$$\text{IF } t = 2 \text{ mm and } d = 65 \text{ mm}$$

$$\eta = S_y / \sigma_x = 3.33$$

$$A = 395.84 \text{ mm}^2$$



APPENDIX 17 (CONTINUED)

MASS OF EACH HOLLOW SECTION

$$\rho A l = 2770 \text{ kg/m}^3 (395.84 \text{ mm}^2) \left(\frac{\text{m}}{1000 \text{ mm}}\right)^2 (0.5 \text{ m})$$
$$= 0.548 \text{ kg}$$

AXIAL LOAD STRESS

$$\sigma_y = \frac{F/2}{\pi/4(d^2 - d_i^2)} = 3.158 \text{ MPa}$$

MUCH LESS THAN σ_y

DEFLECTION BY COMPRESSION

$$I = \frac{\pi}{64}(d^4 - d_i^4)$$
$$= 1.966 \times 10^4 \text{ mm}^4$$

$$P_u = \frac{I C \pi^2 E}{l^2}$$
$$= \frac{(1.966 \times 10^4)(1.2)(\pi)^2(72.345)}{(1500)^2}$$
$$= 673.74 \text{ kN}$$

$$P_u \geq \eta P$$

$$673.74 \geq 3(F/2)$$

$$673.74 \geq 3.75$$

SAFE FROM BUCKLING

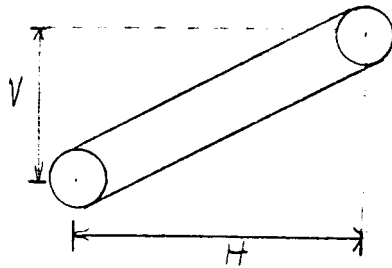
WEIGHT OF TRUSS FOR THE BOOM

$$(1.63 \text{ m/sec}^2) \left(\frac{10 \text{ beams}}{\text{section}} \right) \left(\frac{\text{Mass}}{\text{beam}} \right) \left[\frac{\text{length of truss}}{\text{beam length}} \right]$$
$$= (1.63)(10)(0.548) \left(\frac{5.5}{1.5} \right)$$
$$= 98.457 \text{ N}$$

[COMPARED TO A 180 lb_f MAN
WEIGHING 66N ON THE MOON]

Power for Conveyor

from page 10-47 in Mark's Standard Handbook for Mechanical Engineers the power for a conveyor is given by the following formula:



$$P = (.002H + .001V)CT$$

$P \equiv$ motor horsepower required

$T \equiv$ capacity in tons per hour

$H \equiv$ horizontal run in feet

$V \equiv$ vertical lift in feet

$C \equiv$ constant (varies from 1 to 2.5)

$c=1$ for graphite, clean coal, soybean

$c=1.2$ for beans, slack coal, sand, dust, flour

$c=1.5$ for salt

$c=2$ for cobs, fly ash, sugar, zinc oxide

$c=2.5$ for alum borax, limestone
pulverized

for the miner $V=4.2$ ft, $H=9.4$ ft, $C=$

T , the capacity, is dependent on the speed and the depth of the collection blade; the production rate. If the front wheel rotates at 20 rpm then:

$$\text{speed of vehicle} = 2\pi(3m) \times 20 \text{ rpm} = 37.699 \frac{m}{min}$$

$$= 2261.947 \frac{m}{hr} \text{ assuming no slip}$$

the collection blade is 2 meters wide and soil weighs $587.5 \frac{lb}{m^3}$ on the moon. If the blade depth is:

collection blade depth	6 cm	4 cm	2 cm
volume collected per hour	$271.43 \frac{m^3}{hr}$	$180.95 \frac{m^3}{hr}$	$90.478 \frac{m^3}{hr}$
weight on moon, collected per hour	$159492 \frac{lb}{hr}$	$106329 \frac{lb}{hr}$	$53164 \frac{lb}{hr}$
tons per hour collected	$79.75 \frac{ton}{hr}$	$53.16 \frac{ton}{hr}$	$26.58 \frac{ton}{hr}$
Power required for conveyor	2.20 HP	1.47 HP	0.734 HP

Appendix 19

Selection of Drive Motors

Type: Permanent Magnet, brushless,
Series wound, DC motor
With gear reduction to obtain
20 rpm wheel speed

$$H = Tw$$

where H = power, Watts
 T = torque, N.m
 w = angular velocity, rad/sec.

front wheel

$$\text{ground speed} = .733 \text{ m/sec}$$

$$\text{angular velocity} = 2.094 \text{ rad/sec. (20 rpm)}$$

$$\text{torque required} = 1196 \text{ N.m}$$

$$\begin{aligned}\text{power, } H &= 1196 (\text{N.m}) \cdot 2.094 (\text{rad/sec}) \\ &= 2,505 \text{ Watts} \\ &= 3.36 \text{ Hp}\end{aligned}$$

5 hp motor will be used

rear wheel

$$\text{angular velocity} = 2.094 \text{ rad/sec (20 rpm)}$$

$$\text{torque required} = 503 \text{ N.m}$$

$$\begin{aligned}\text{power } H &= 503 (\text{N.m}) \cdot 2.094 (\text{rad/sec}) \\ &= 1053 \text{ Watts} \\ &= 1.4 \text{ Hp}\end{aligned}$$

3 hp motor will be used

Production for Miner

- the collection blade is 2m wide
- the speed is $2261.9 \frac{m}{hr}$ using a rotational speed of 20 rpm for the front wheel and assuming no slip
- the volume collected per hour is dependent on the collection blade depth;

Blade depth:	6cm	4cm	2cm
Volume collected per hour:	$271.43 \frac{m^3}{hr}$	$180.95 \frac{m^3}{hr}$	$90.48 \frac{m^3}{hr}$

$$\text{volume collected per unit time} = \text{width} \times \text{depth} \times \text{speed}$$

Appendix 21

Stress Calculations on Blades

For Titanium alloy (Ti-6Al-4V)

$$E = 1.1 \times 10^{11} \text{ Pa}$$

$$\rho = 4.47 \text{ g/cm}^3$$

$$\sigma_y = 8.27 \times 10^8 \text{ Pa @ Room temp.} \quad \therefore S_{sy} = 3.31 \times 10^8 \text{ Pa}$$
$$= 6.62 \times 10^8 \text{ Pa @ } 300^\circ\text{F}$$

tensile:

$$\sigma = \frac{Mc}{I} \quad I = \frac{1}{12}bh^3 = \frac{1}{12}(2\text{m} \times .01\text{m})^3 = 1.67 \times 10^{-7} \text{ m}^4$$

$$\sigma = \frac{(1271\text{N})(.1156\text{m})(.005\text{m})}{1.67 \times 10^{-7} \text{ m}^4} = \underline{4.4 \times 10^6 \text{ Pa}}$$

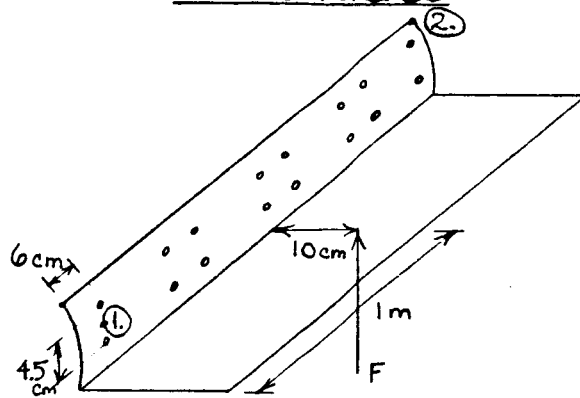
$$n = \frac{\sigma}{\sigma_y} = \underline{150}$$

shear:

$$\tau = \frac{4}{3} \frac{F}{A} = \frac{4}{3}(1271\text{N})/[(.01\text{m} \times 2\text{m})] = 8.5 \times 10^4 \text{ Pa}$$

$$n = \frac{3.31 \times 10^8}{8.5 \times 10^4} = \underline{3894}$$

Stress Calculations for Bolts on Blades



For Ti-6Al-4V:

$$S_{sy} = \frac{1}{2}(6.62 \times 10^8 \text{ Pa})$$

$$= \underline{3.31 \times 10^8 \text{ Pa}}$$

I. Shear:

Worst case: $F = 1300 \text{ N}$

$$F' = \frac{1300 \text{ N}}{16} = 81.25 \text{ N}$$

Bolts at each end are worst-case:

$$\text{at } \textcircled{1}: M = (1300 \text{ N})(.94 \text{ m}) = 1222 \text{ N}\cdot\text{m}$$

$$F'' = \frac{Mr}{4r^2} = \frac{(1222)(.0225 \text{ m})}{4(.0225 \text{ m})^2} = 1.36 \times 10^4 \text{ N}$$

$$V = \sqrt{F'^2 + F''^2}$$

$$= \sqrt{(81.25)^2 + (1.36 \times 10^4)^2} = 1.36 \times 10^4 \text{ N}$$

$$\tau = \frac{1.36 \times 10^4 \text{ N}}{\pi(.007)^2} = \underline{8.83 \times 10^7 \text{ Pa}}$$

M14 x 1.5 bolts \rightarrow

$$n = \frac{S_{sy}}{\tau} = \underline{3.75}$$

II. Tensile:

Sum moments about pt. $\textcircled{2}$:

$$1300(.1 \text{ m}) = P(.07)$$

$$P = 1857 \text{ N}$$

Tensile force on eight bolts (Those subjected to tensile loads)

$$P = \frac{1857}{8} = \underline{232 \text{ N}}$$

$$\sigma = \frac{232 \text{ N}}{\pi(.007)^2} = \underline{1.5 \times 10^6 \text{ Pa}}$$

$$n = \frac{6.62 \times 10^8}{1.5 \times 10^6} = \underline{441}$$

Appendix 23

Power Calculations for Wheels

Using figure _____ : $\phi = 36^\circ$ $R = .796$
 $\delta = 24^\circ$

For first blade: $\beta = 30^\circ$

$$\text{moment arm} = .35 + 15 - \frac{1}{3}(.15 + .35(\cos(30^\circ) - 1)) \\ = \underline{.4656 \text{ m}}$$

$$\frac{\beta}{\phi} = +.83$$

$$\text{from chart } K_p = 33.5 \times R = 26.7$$

$$P_p = (26.7) \left(1.5 \frac{\text{g}}{\text{cm}^3} \right) \left(\frac{1000 \text{ cm}^3}{1 \text{ m}^3} \right) \left(\frac{9.81 \text{ m}}{6 \text{ s}^2} \right) \left(\frac{1 \text{ kg}}{1000 \text{ g}} \right) (H^2) \\ = (26.7) \times (2453) \times (.1)^2 = 696 \text{ N/m} \times 2 \text{ m} \\ = 1392 \text{ N}$$

$$\text{Torque}_1 = \underline{592 \text{ N-m}}$$

$$P = \underline{1271 \text{ N}}$$

For second blade: $\beta = 0^\circ$

$$\text{moment arm} = .35 + .10 = .45 \text{ m}$$

$$\frac{\beta}{\phi} = 0.$$

$$\text{from chart } K_p = 11.5 \times R = 9.154$$

$$P_p = (9.154)(2453)(.15)^2 = 505 \text{ N/m} \times 2 \text{ m} = 1010 \text{ N}$$

$$P = \underline{923 \text{ N}}$$

$$\text{Torque}_2 = \underline{415 \text{ N-m}}$$

For third blade :

$$W = (.2129 \text{ m}^3)(2000 \text{ kg/m}^3) \left(\frac{9.81 \text{ m}}{6 \text{ s}^2} \right) = 696 \text{ N}$$

$$W_t = W \sin 25^\circ = 294 \text{ N}$$

$$F_f = (W \cos 25^\circ)(.2) = 126 \text{ N}$$

$$\text{Torque}_3 = (294 + 126) \times (.45) = 189 \text{ N-m}$$

Appendix 23 continued

Power (con't)

$$\text{Torque} = 592 + 415 + 189 = \underline{1196} \text{ N-m}$$

$$\begin{aligned} \text{Power} &= \text{Torque} \times \omega = (1196 \text{ N-m}) \times (2.1 \text{ rad/s}) \\ &= \underline{2512 \text{ W}} = 3.37 \text{ hp.} \end{aligned}$$

Rear wheel:

$$(\text{Torque}_1 + \text{Torque}_2) / 2 = \underline{503 \text{ N-m}}$$

$$\text{Power} = (503 \text{ N-m})(2.1 \text{ rad/s}) = \underline{1056 \text{ W}} = 1.42 \text{ hp}$$

Appendix 24

POWER CALCULATION FOR STEERING

- Based on power to turn one blade (on rear wheel) through 360° in soil at 12 rpm.

For force on one blade at 90° to soil surface see "Power Calculations for wheels."

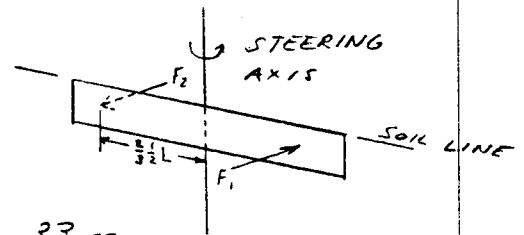
REAR BLADE WIDTH = 1m

$$F_1 = F_2 = (1271\text{N} + 923\text{N})\left(\frac{1}{2}\right) = 1097\text{N}$$

$$\text{Moment arm} = \frac{2}{3}\left(\frac{1}{2}L\right) = \frac{1}{3}L = .33\text{m}$$

$$\text{Torque} = (1097\text{N})(.33\text{m})(2) = 724\text{ N-m}$$

↑
EACH SIDE OF REAR
BLADE ROTATING ABOUT
STEERING AXIS



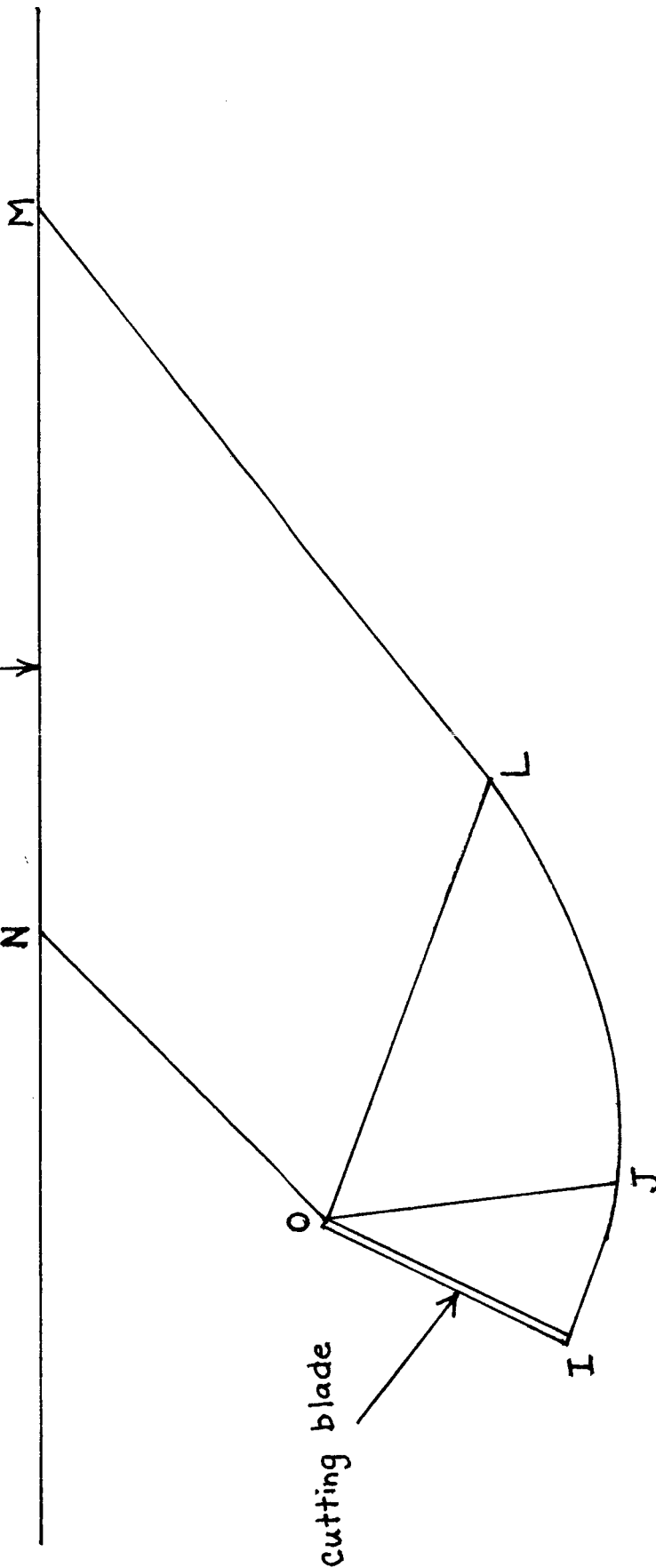
$$\text{POWER} = (724\text{ N-m})(1.26\text{ rad/sec}) = \underline{\underline{912\text{ W}}}$$

APPENDIX 25

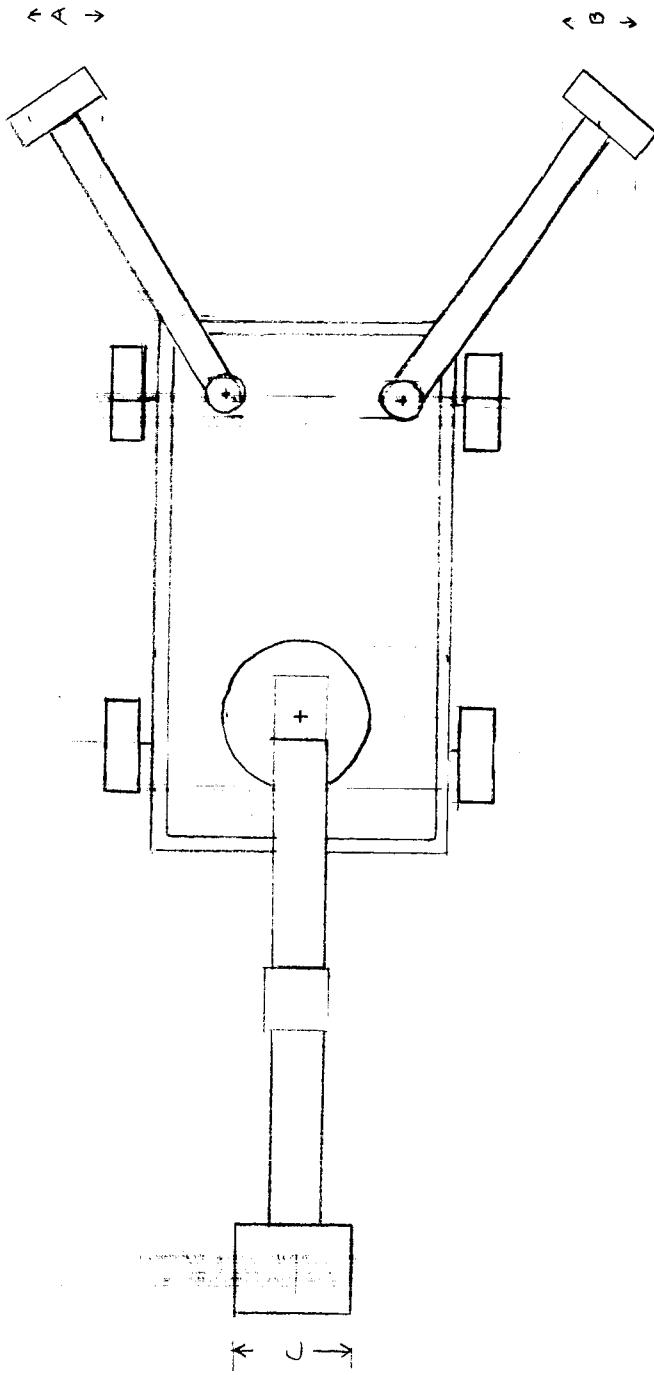
Figures

- 1) Ohde's graphical method
- 2) Sub-surface blade
- 3) Soil force chart
- 4) Proposal # 1 (digging, rejected)
- 5) Proposal # 1 (mining, rejected)
- 6) Proposal # 2 (mining, rejected)
- 7) Proposal # 2 (mining, rejected)
- 8) Proposal # 3 (mining, accepted)
- 9)
 - a - Boom
 - b - Detail of spade
 - c - Detail of spade
 - d - Detail of spade
 - e - Yoke
 - f - Assembly
 - g - Detail of power screws on arm
 - h - Detail of power screws on arm
 - i - Detail of power screws on arm
- 10) Conveyor
- 11) Blade actuator
- 12) Collection blade
- 13)
 - a - Side of wheel cutting soil
 - b - Detail of blade
- 14) Steering
- 15) Assembly
- 16) Front wheel
- 17) Wheel blade

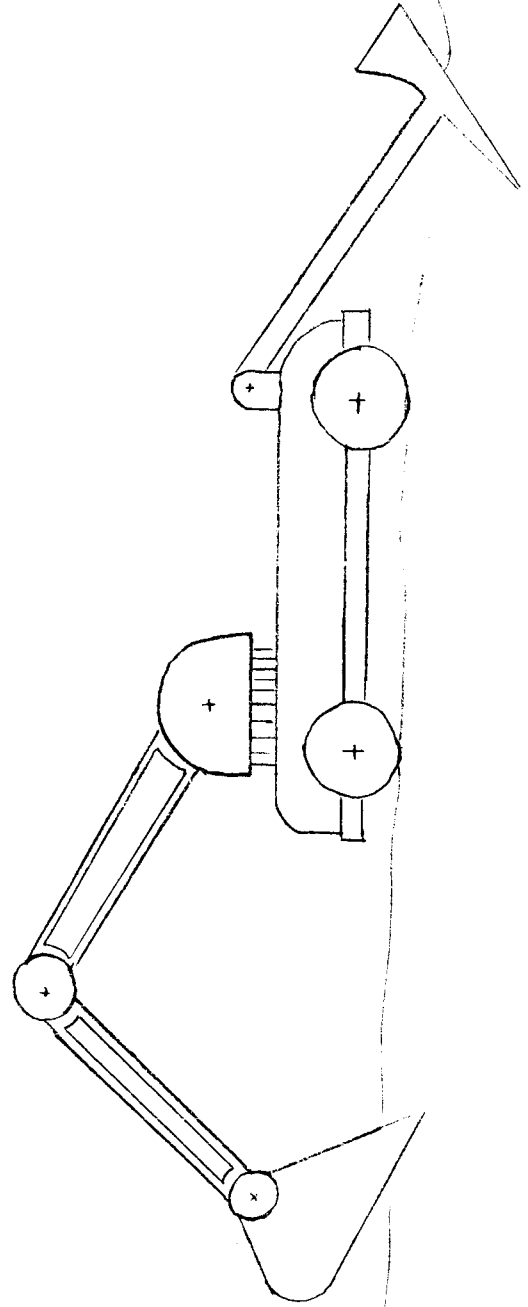
Soil surface



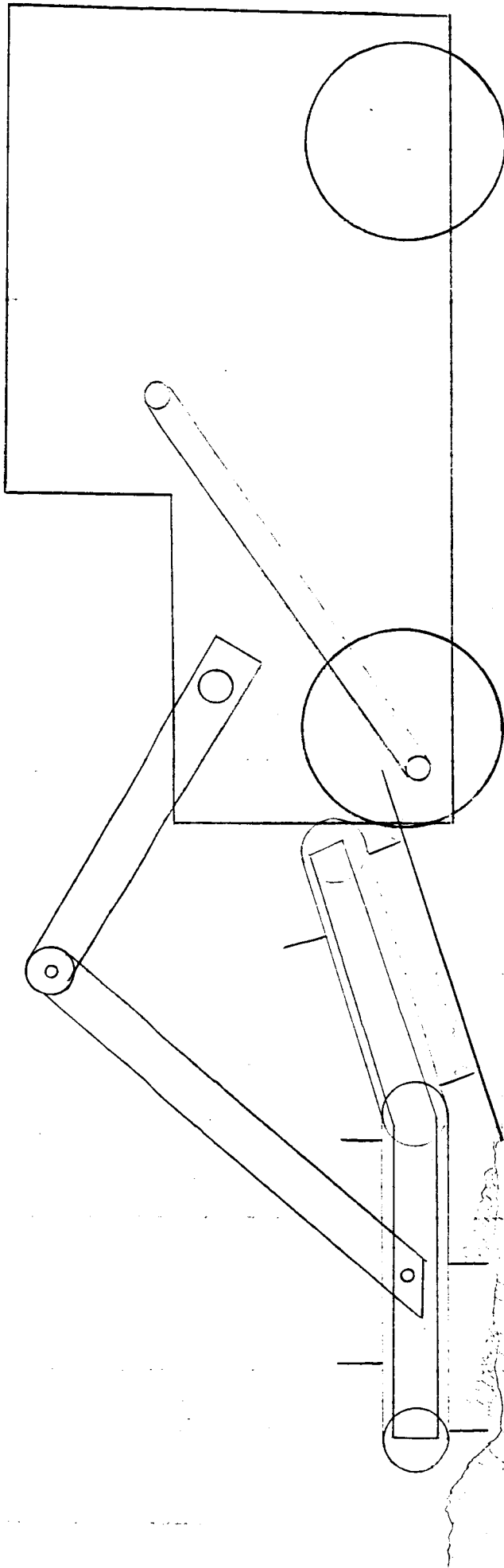
Subsurface cutting blade		
Russ Anderton	# 2	
Figure	5-29-86	



THE FRONT - FACING
AREA OF THE
FIRE-PLOWS (A+B)
MUST BE LARGER
THAN THE AREA
OF THE BUCKET (C)

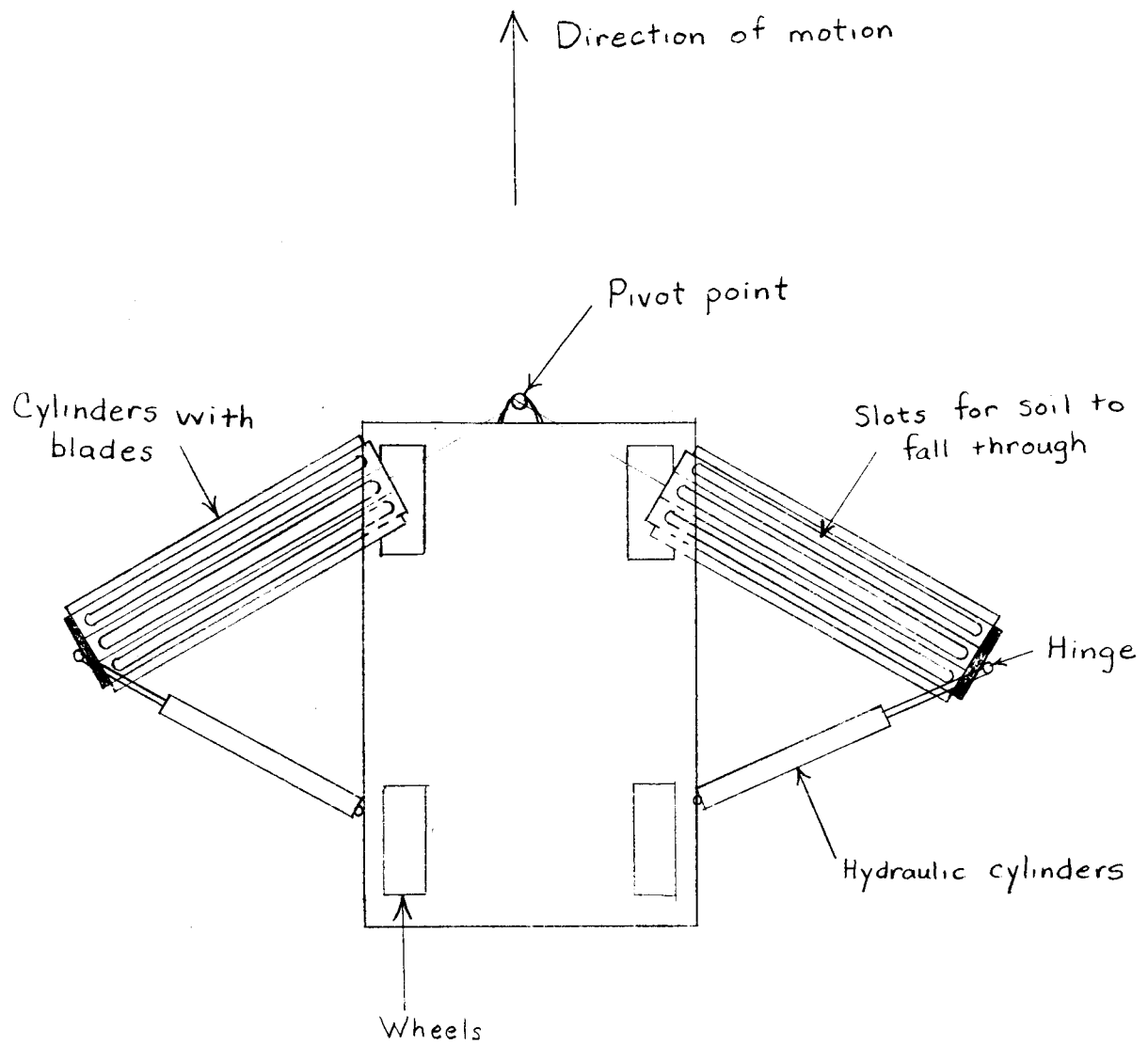


TITLE: PROPOSAL 1 (REJECTED)	
DRAWN BY: JEFF KEESEE	NO: 4
DATE: 6-2-86	



TITLE: MINING PROPOSAL NO. 1	
DRAWN BY: T. ALDRIDGE	NO: 5
SCALE:	DATE: 5-31-86

Proposal #2



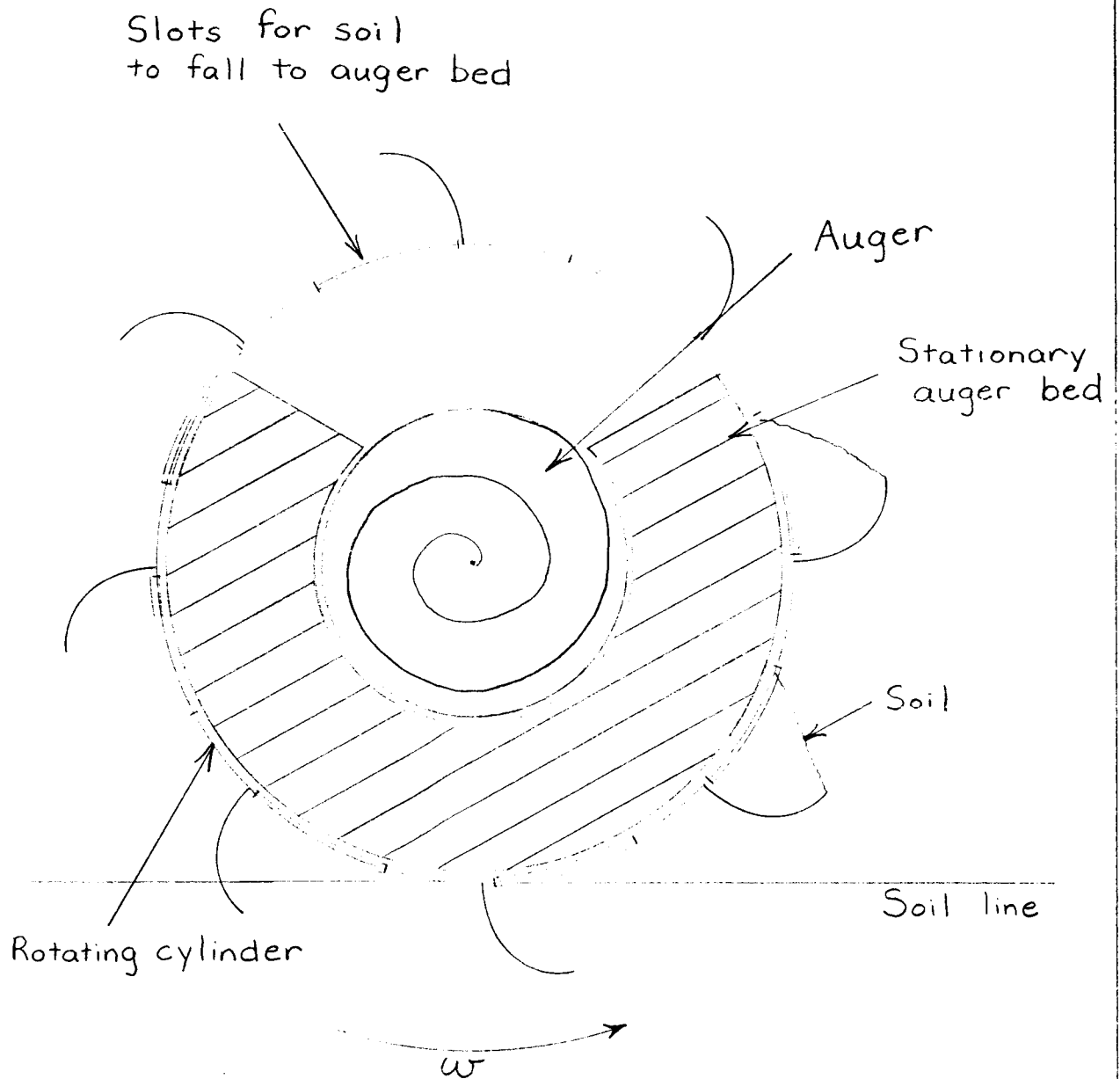
Top view of vehicle

Russ Anderton #6

Figure

5-29-86

Proposal #2

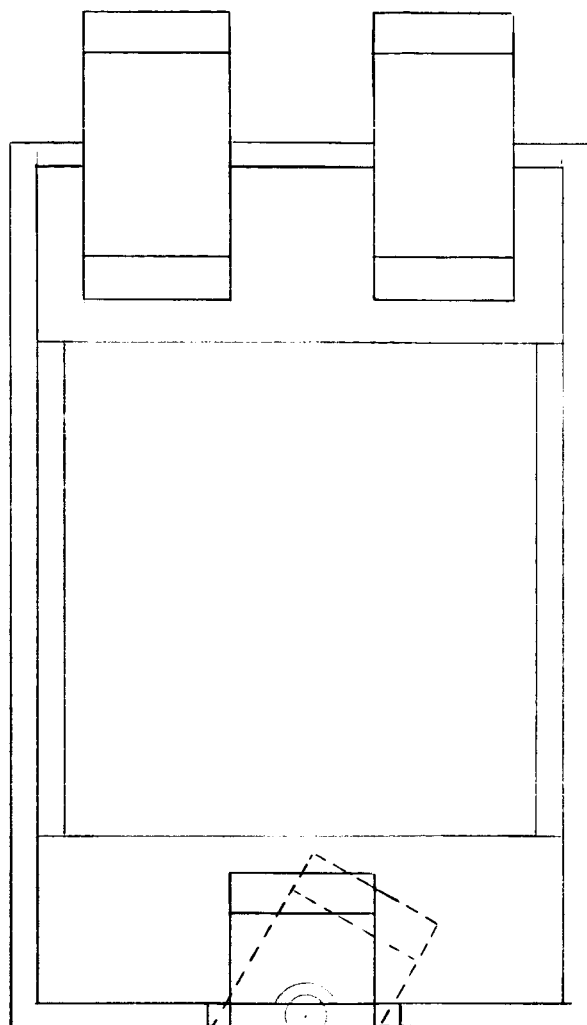


Side view of wheel

Russ Anderton # 7

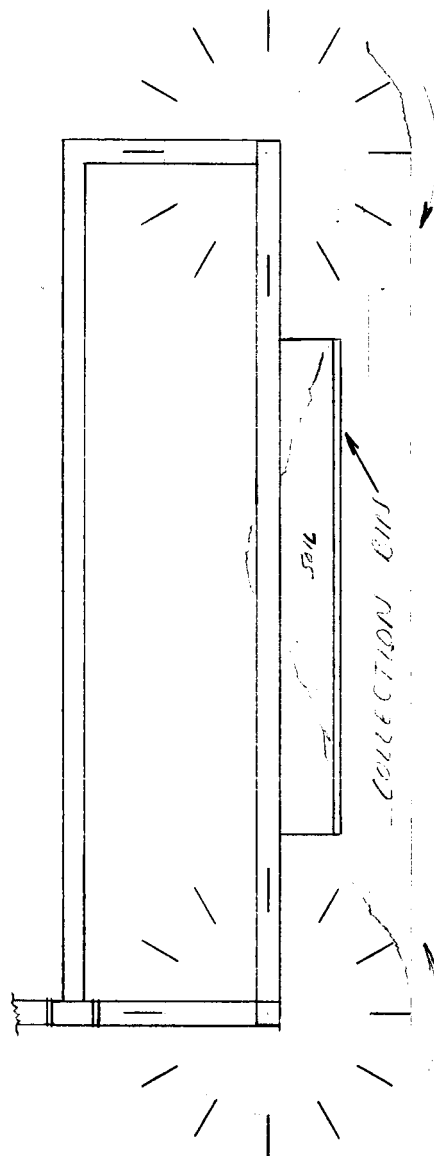
Figure

5-29-86



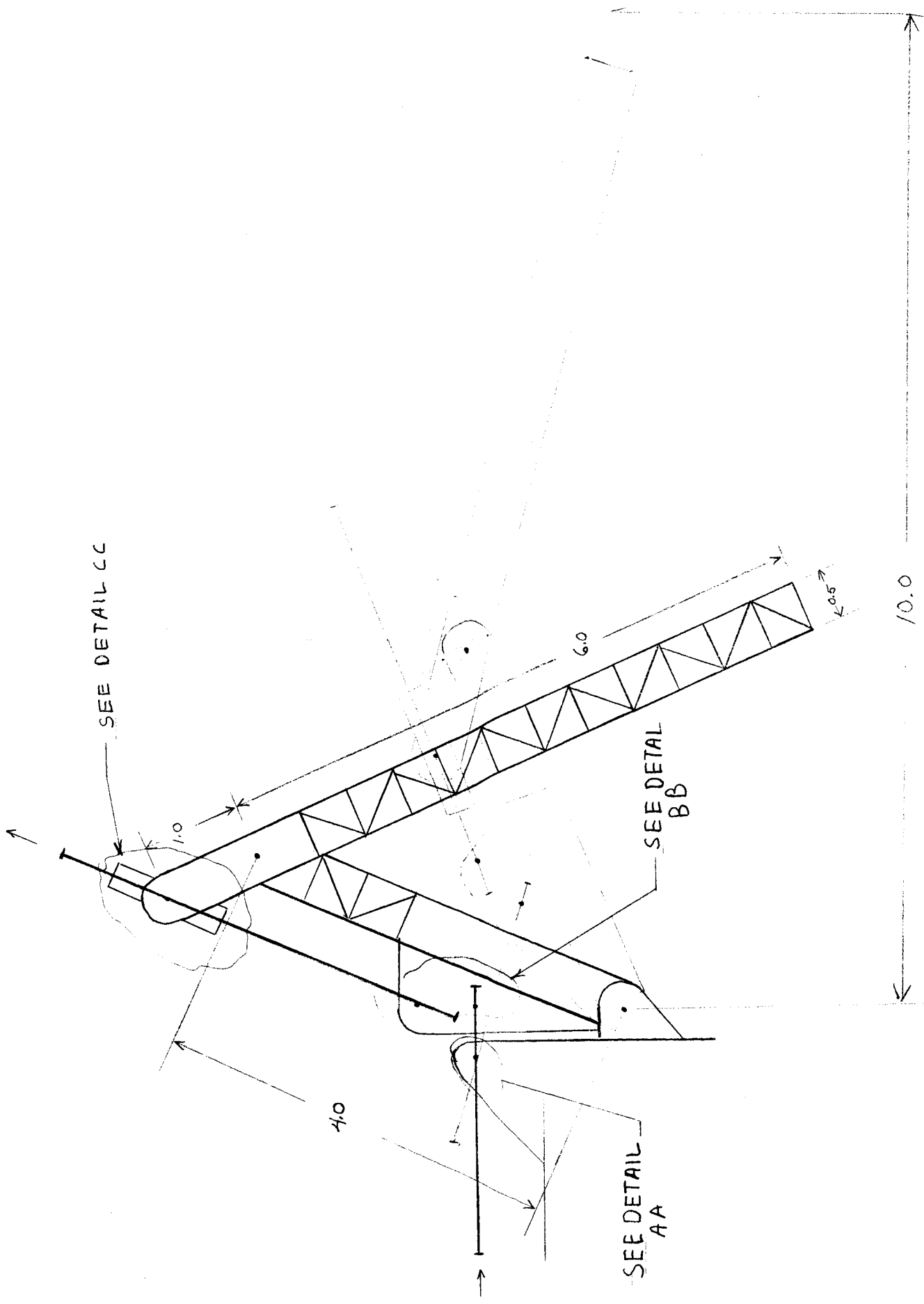
REAR STACKING

TRAVEL



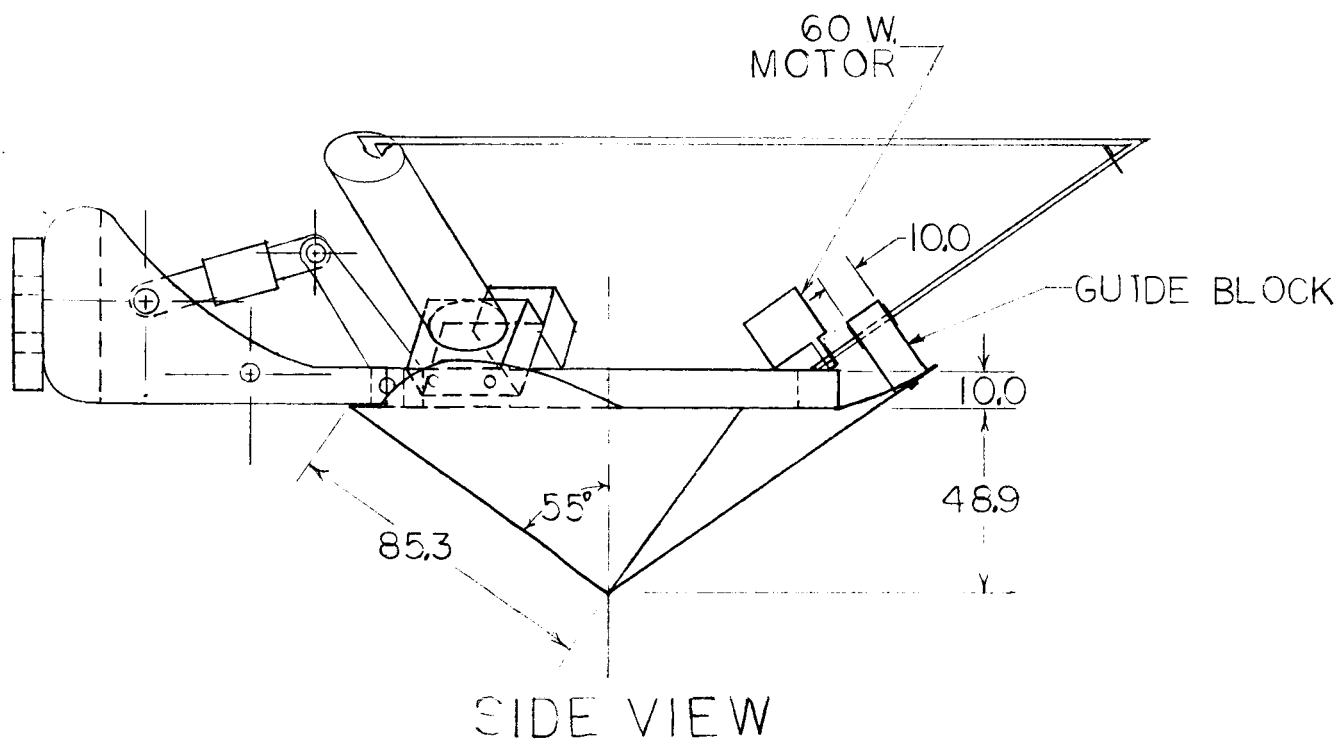
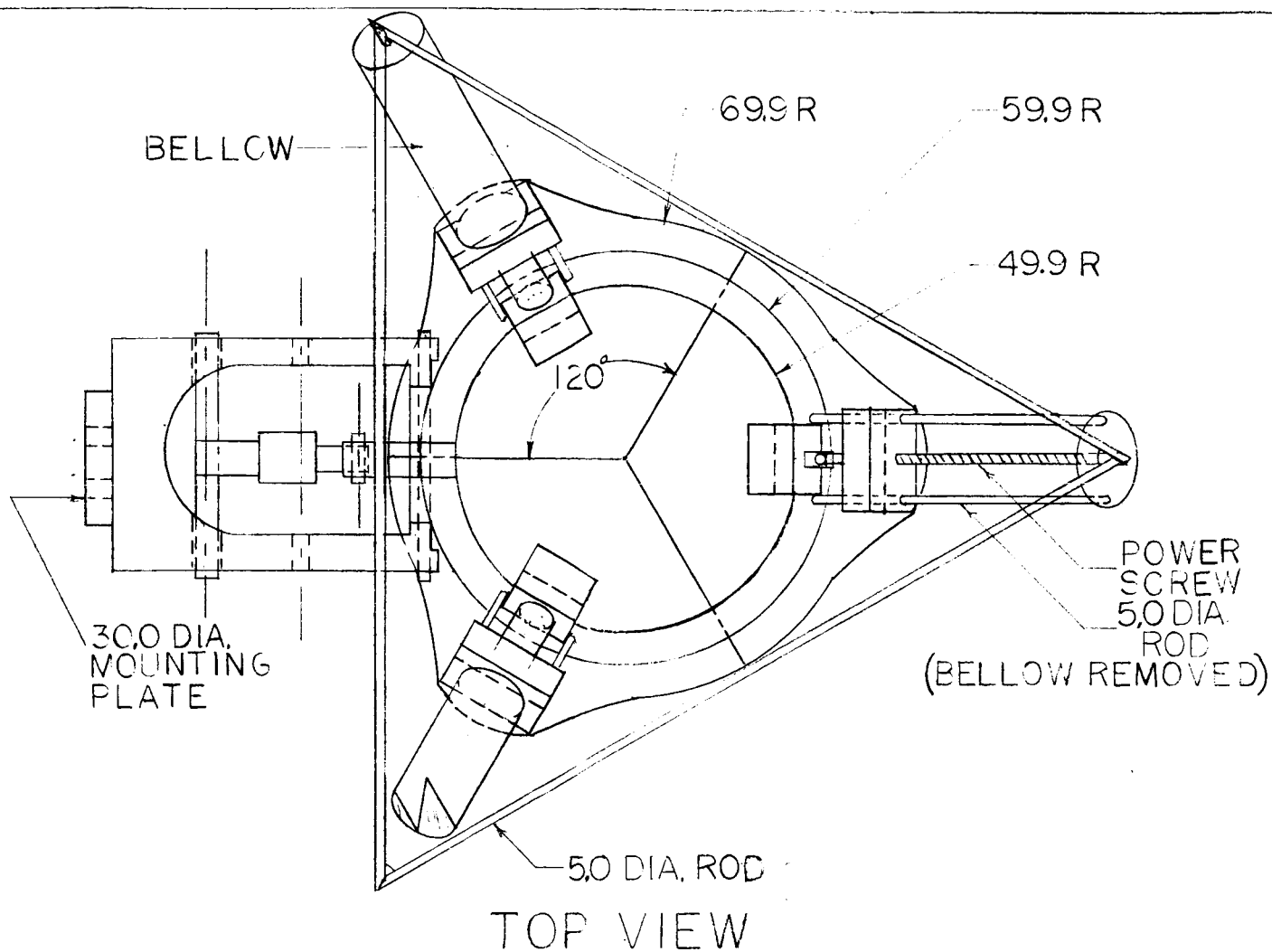
COLLECTION CHUTE

TITLE: MINING PROADER 3	NO: 8	DATE: 29/11/56
DRAWN BY: RALLEN		
SCALE:		



TITLE: BOOM		NO: 9A	
DRAWN BY: JEFF KEESEE		DATE: 6-2-86	
SCALE:			





NOTES: ALL DIMENSIONS IN CM./
SEE DRAWINGS "E" AND "C" FOR
YOKE AND ACTUATOR DETAILS

TITLE: SPADE BUCKET (1 OF 2)	
DRAWN BY: BEN KUCHEL	NO: 9B
SCALE: 1CM=20CM.	DATE: 5-29-86

40.0 DIA. BASE
W/ BEARING MOUNTS

5.0 DIA.

1.27 DIA. BALL
BEARING SCREW

ELECTRIC MOTOR
60 WATT, 1608 RPM

BEARING
ASSEMBLY

NOTE: TOTAL LENGTH OF
SCREW & GUIDES, FROM
BEARING-TO-BEARING
IS 110.0 CM.

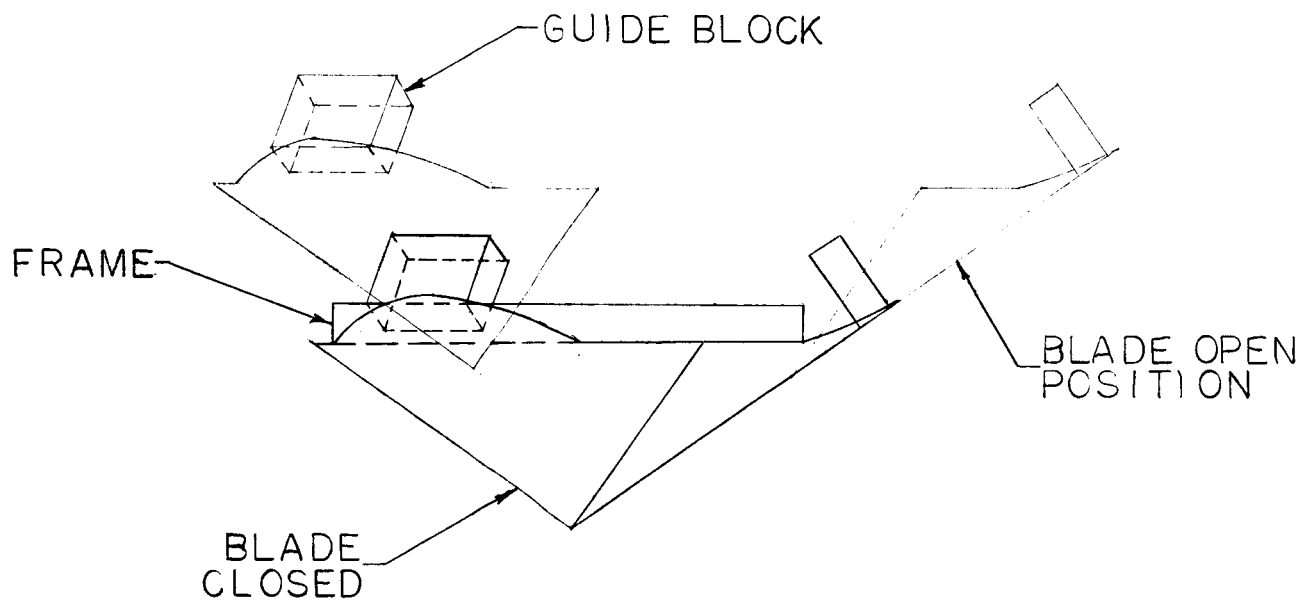
FRAME & BEARING
MOUNTS REMOVED

NOTE: ALL DIM. IN CM. / BELLOWS REMOVED FOR CLARITY

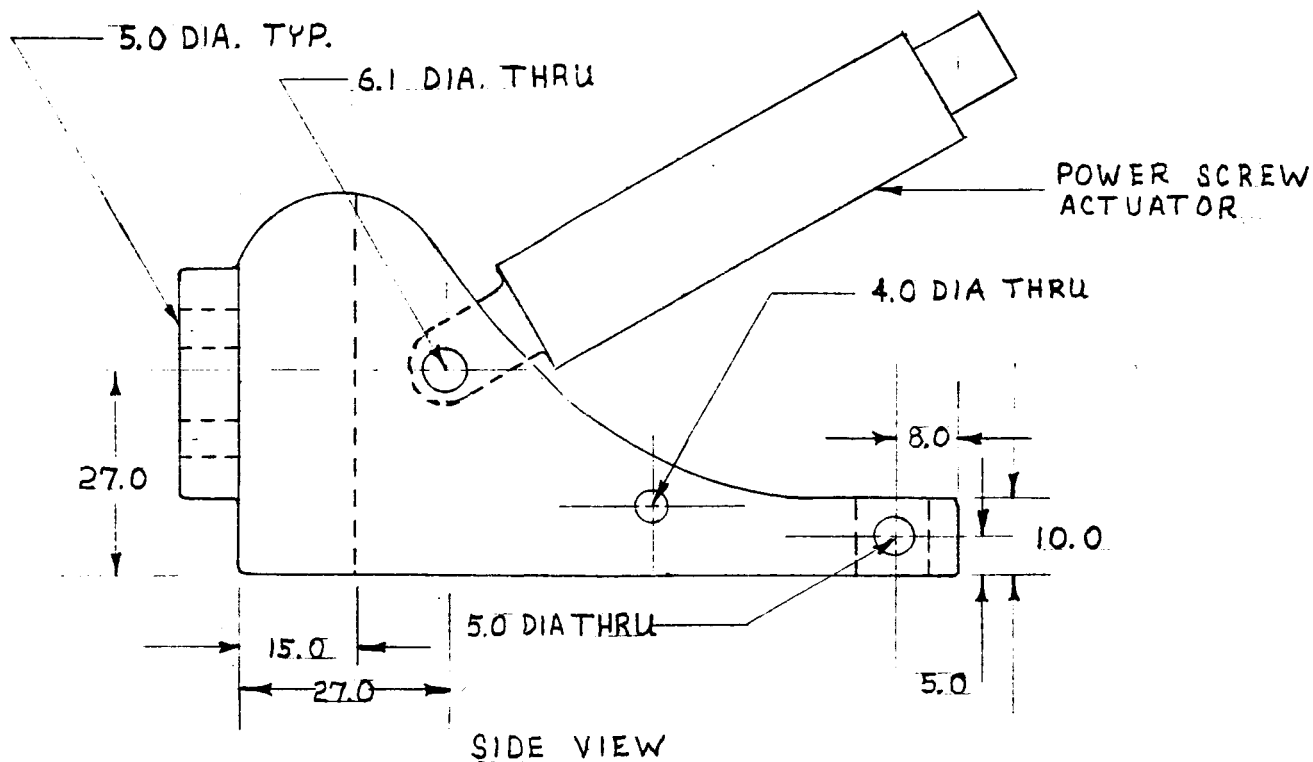
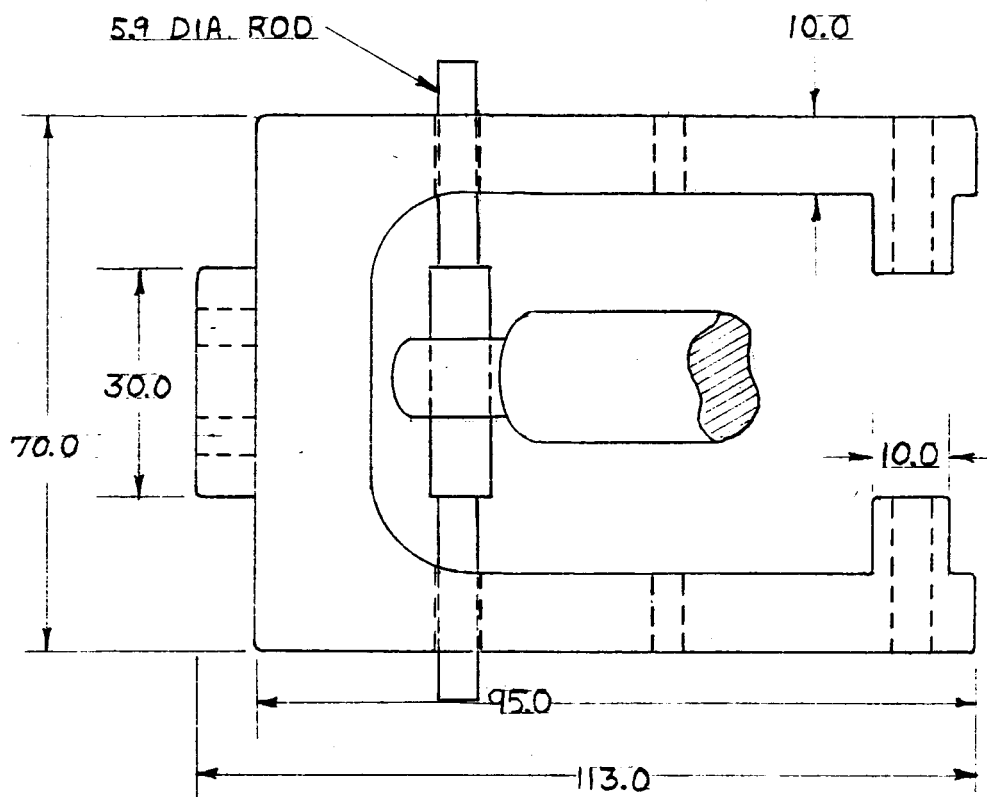
TITLE: BLADE ACTUATOR (1 OF 3)

DRAWN BY: BEN KUCHEL NO. 9C

SCALE: 1 CM. = 10 CM. DATE: 5-29-80



TITLE:WORKING BUCKET	
DRAWN BY:BEN KUCHEL	NO. 9D
SCALE:1 CM=20 CM	DATE:5-29-86



NOTE:

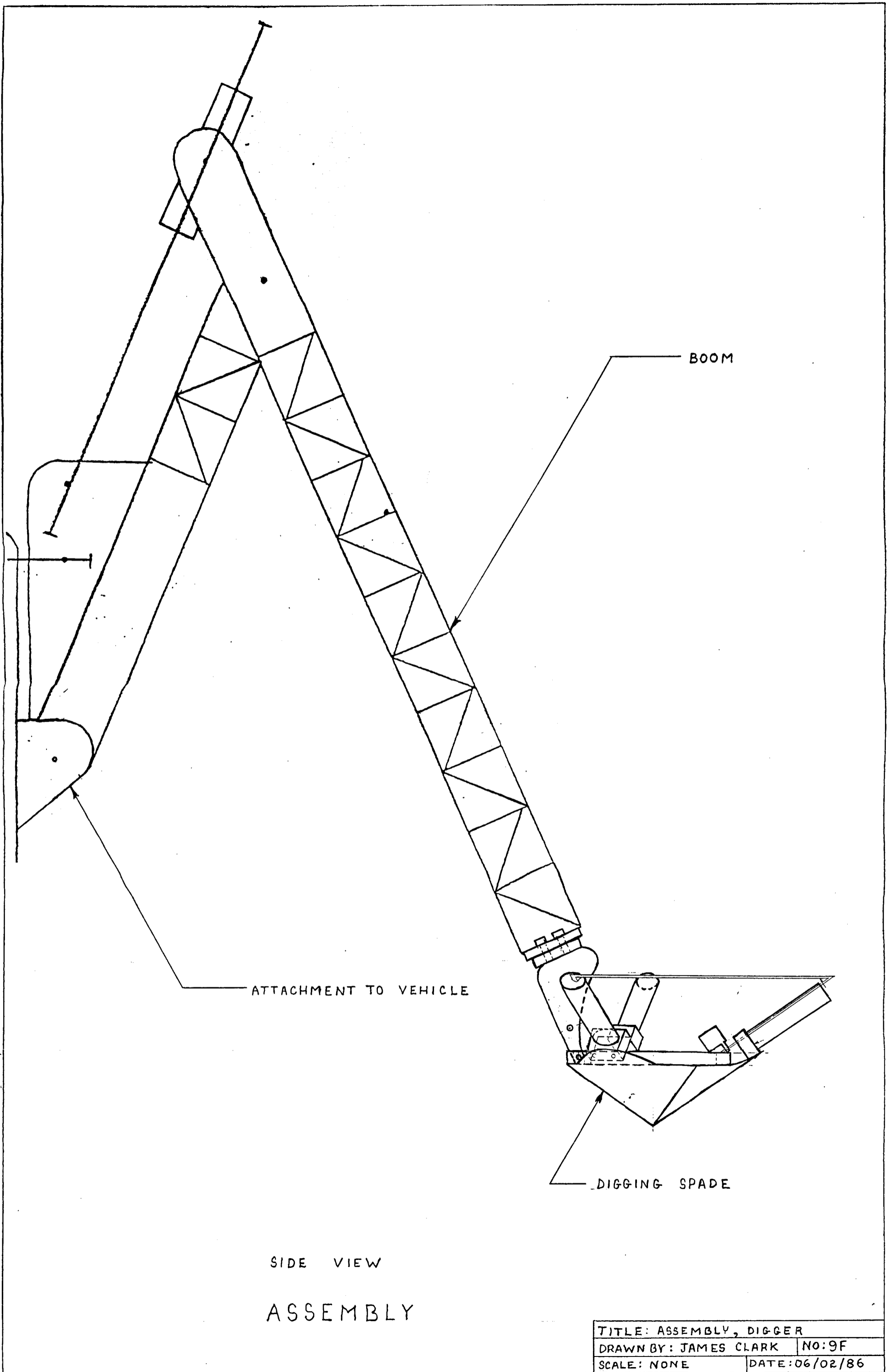
1. ALL DIM. IN CM.

TITLE: YOKE

DRAWN: JAMES CLARK NO: 9E

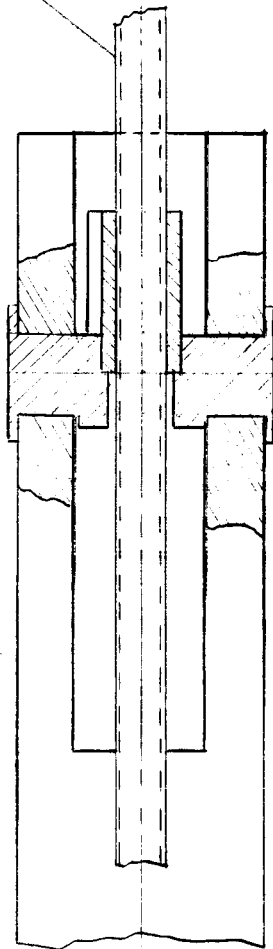
SCALE: 1/10

DATE: 05/30/86



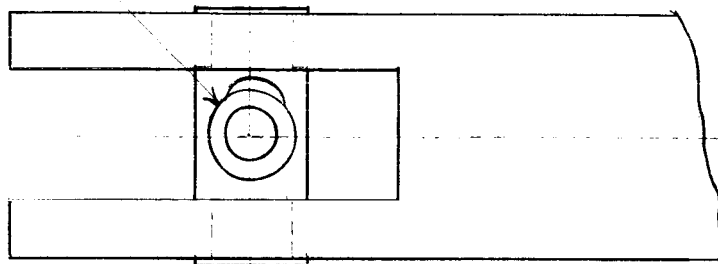
TITLE: ASSEMBLY, DIGGER	
DRAWN BY: JAMES CLARK	NO: 9F
SCALE: NONE	DATE: 06/02/86

5.7 DIA.
1.28 LEAD

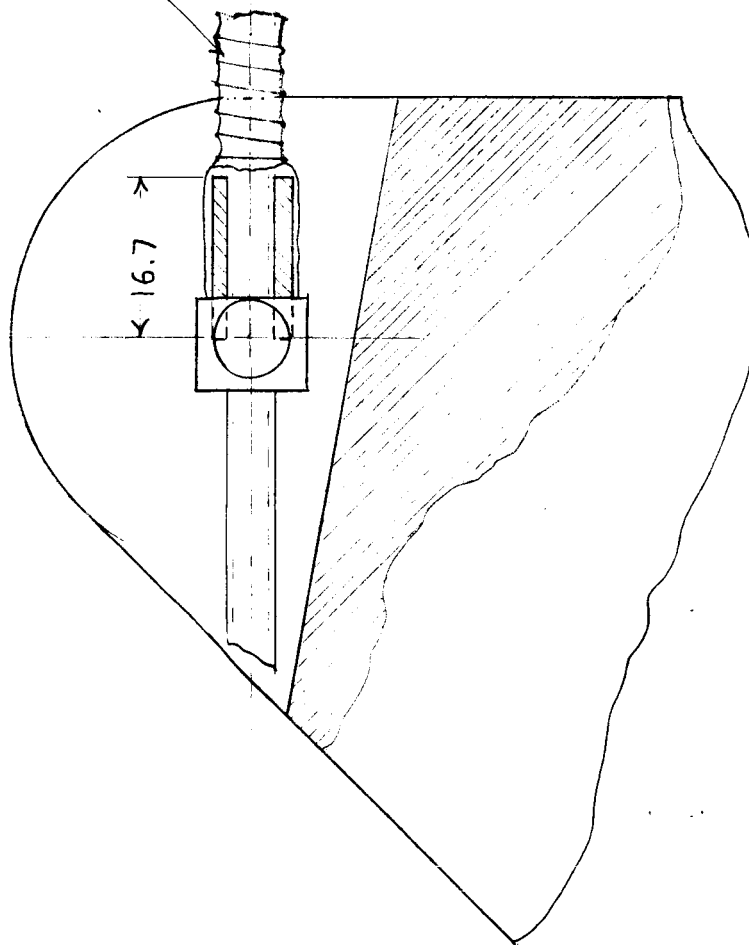


ALL DIMENSIONS
IN cm

8.6 DIA.



DUST
COVER
OVER ALL
SCREWS



16.7

DETAIL AA

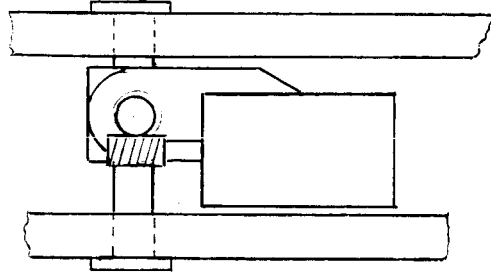
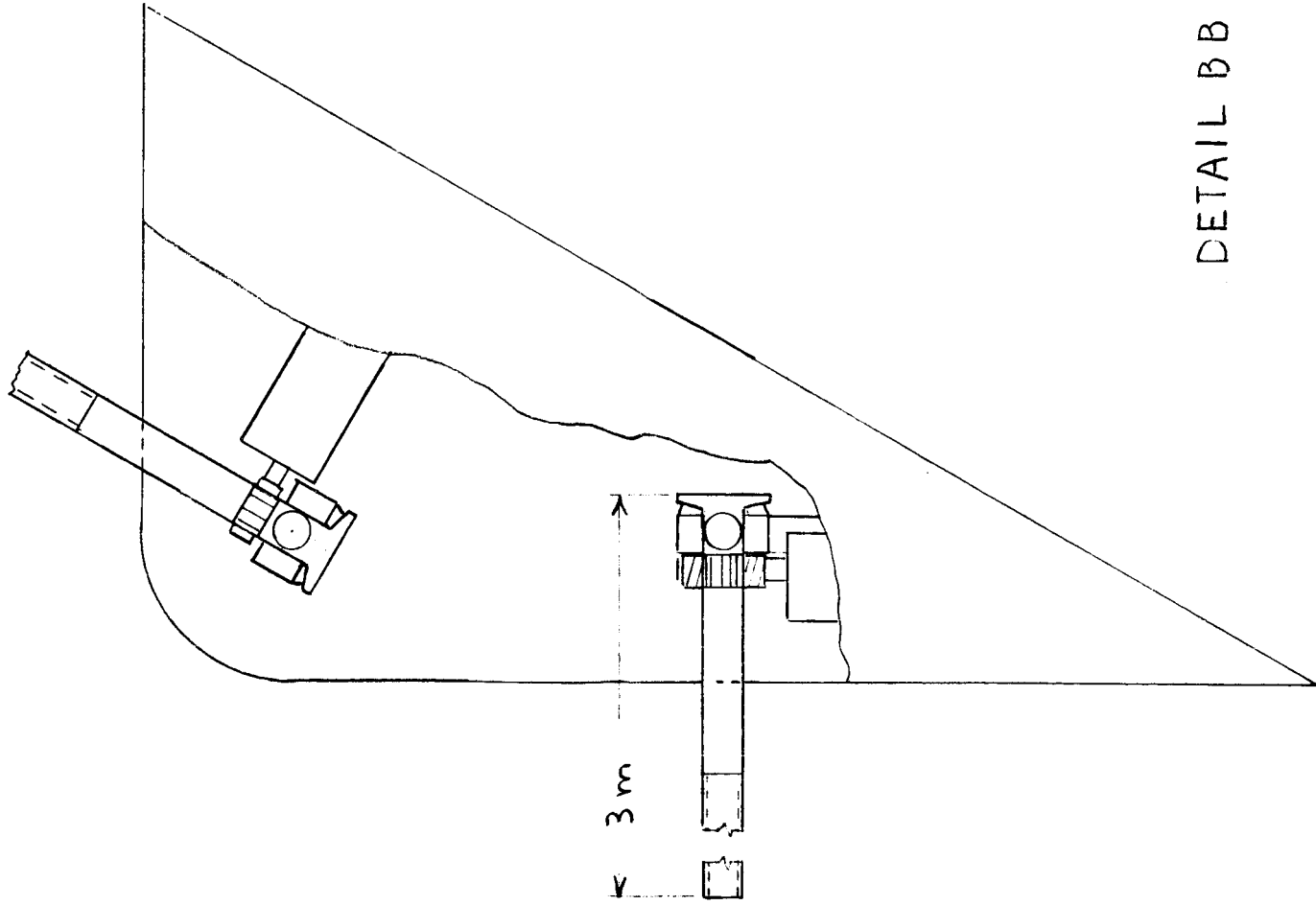
Title: Power Screw Detail

Drawn By: Dong Peace

No: 96

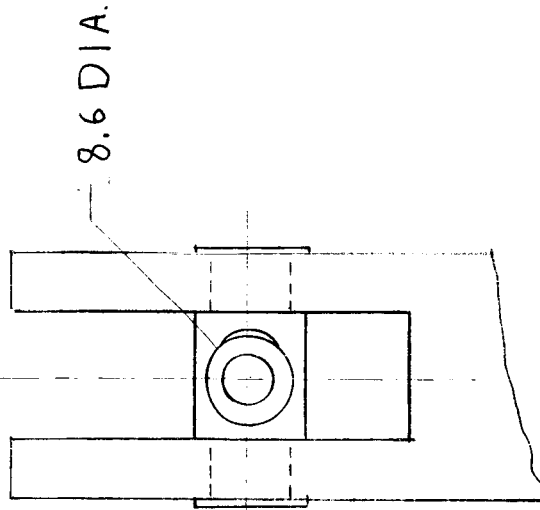
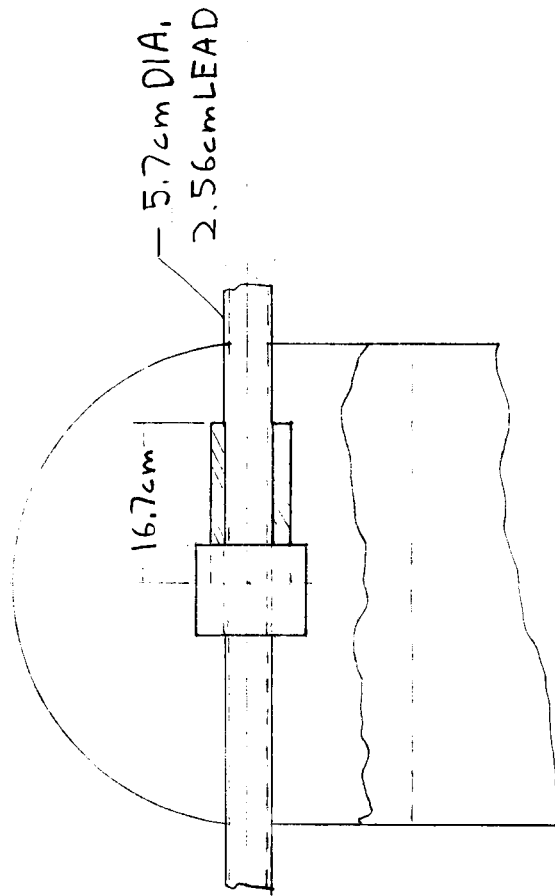
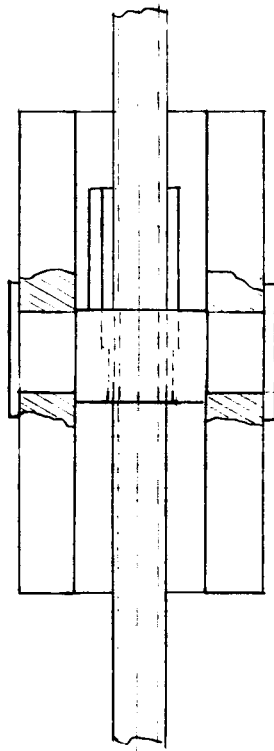
Scale: 1cm = 8cm

Date: 6-2-86

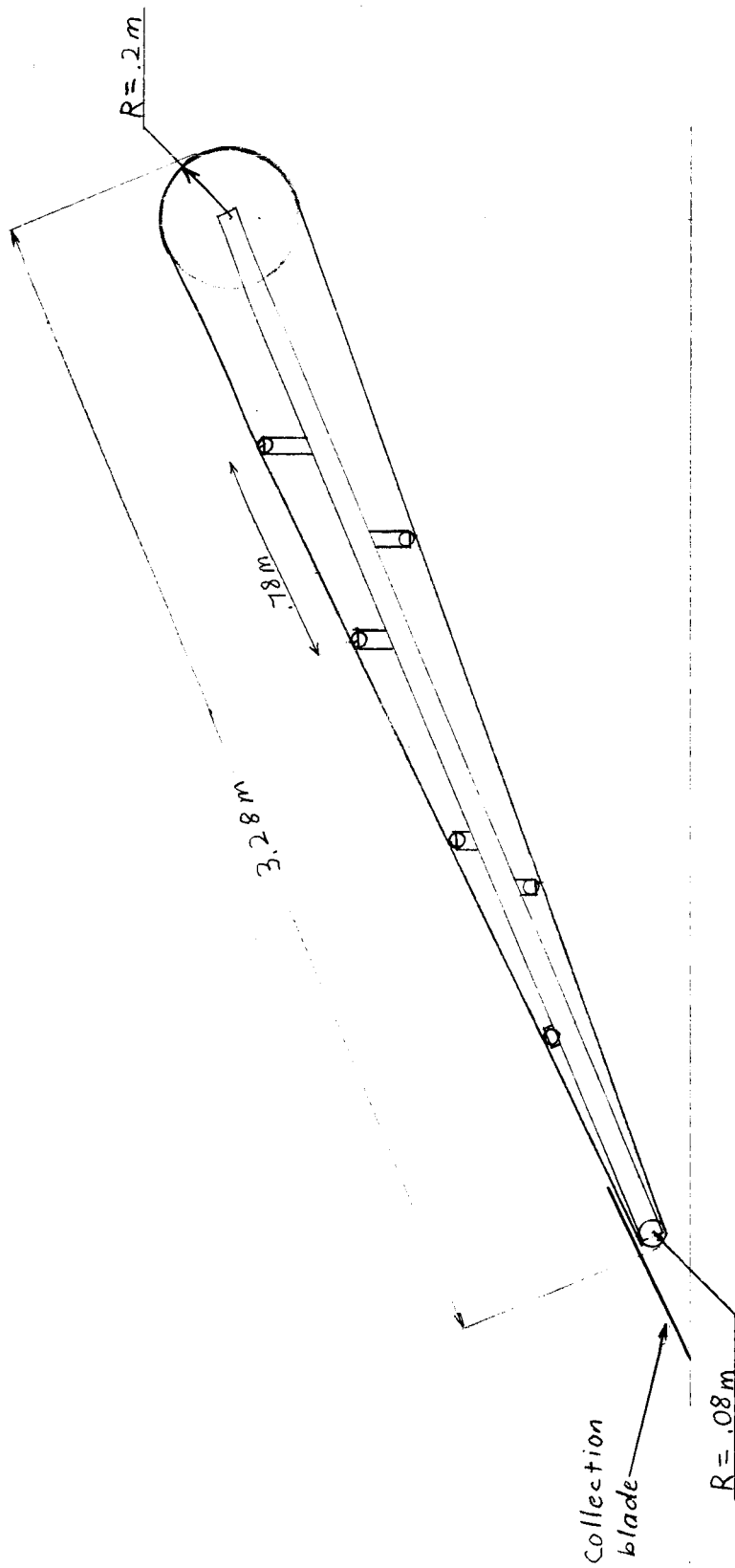


DETAIL BB

Title: Power Screw Detail		
Drawn By: Doug Peace	No: 914	
Scale 1cm = 10cm	Date: 6-2-86	



Title: Power Screw Detail		
Drawn By: Dong Peace	No: 91	
Scale: 1cm = 8cm	Date: 6-2-86	



TITLE: CONVEYOR

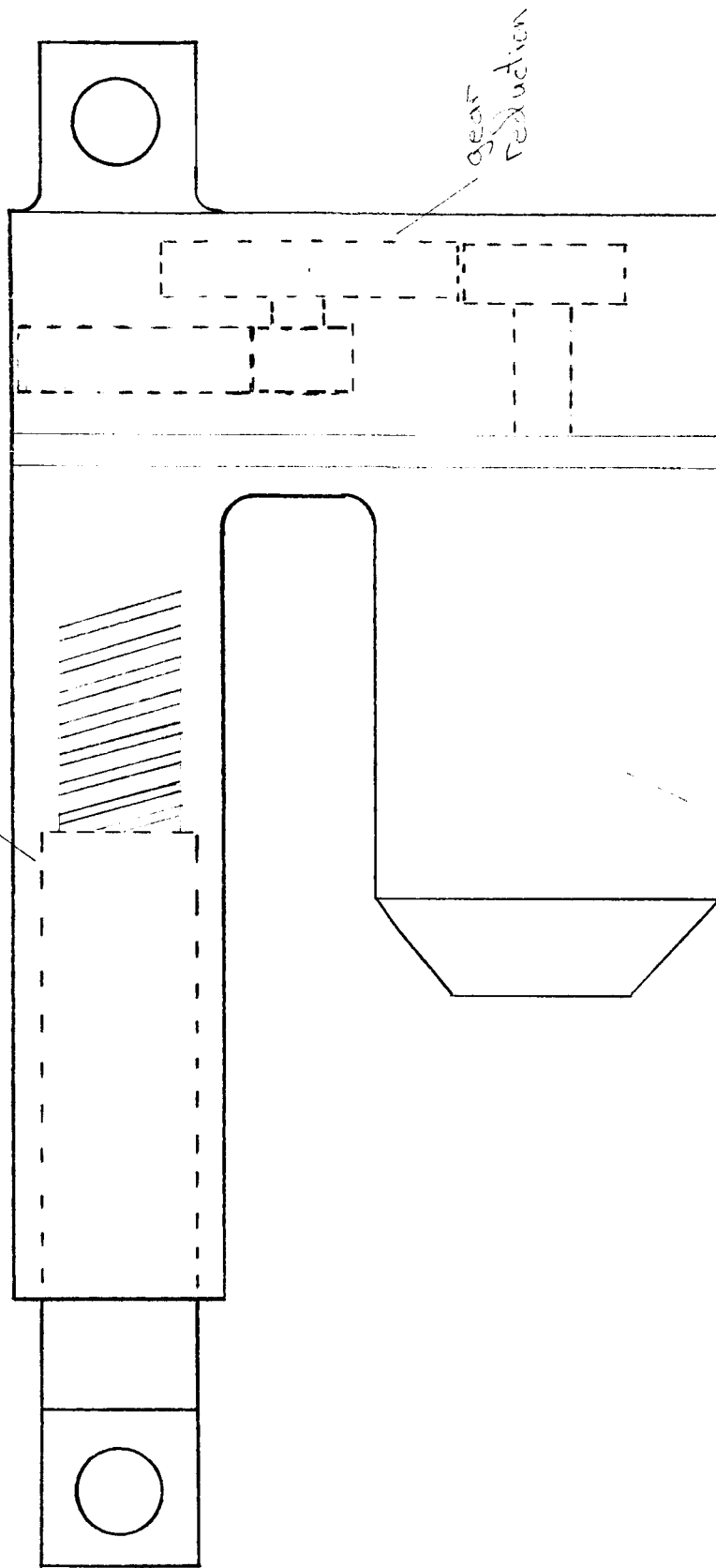
DRAWN BY: T. ALDRIDGE

NO: 10

SCALE: 1 cm = .2 m

DATE: 6-1-86

Ball Bearing Screw

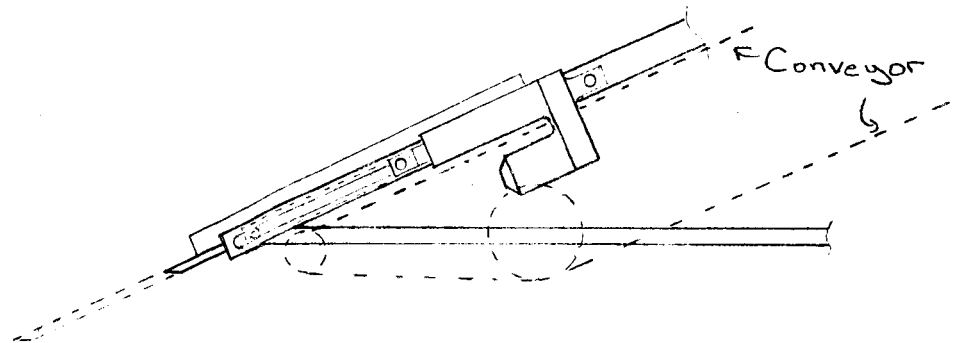
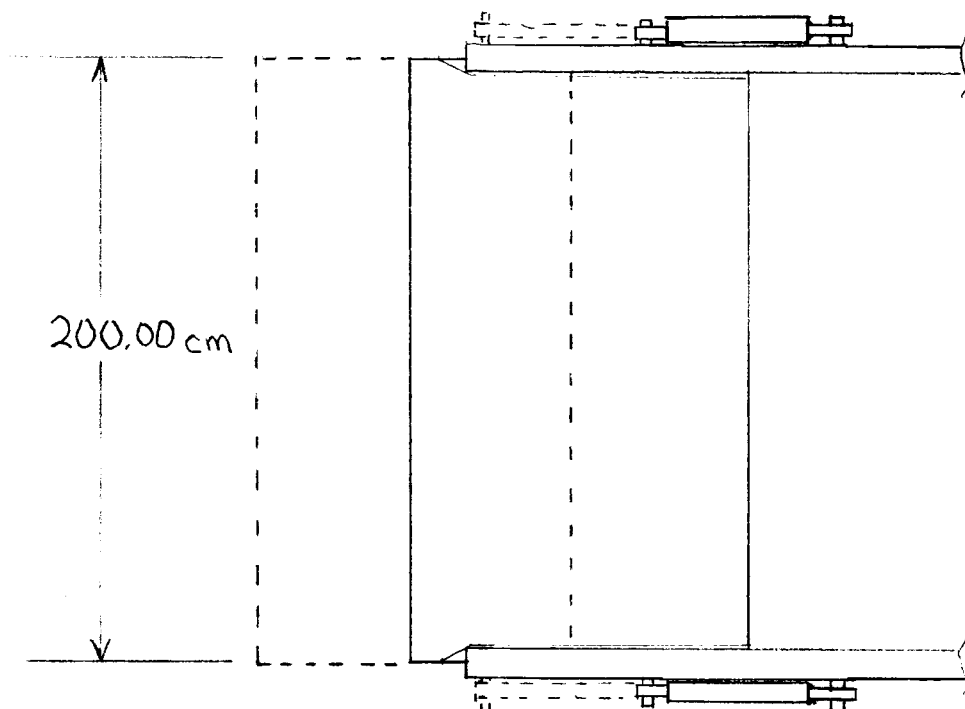


DC motor

Blade Actuator 1.82

Drawn by DPC

No. 11

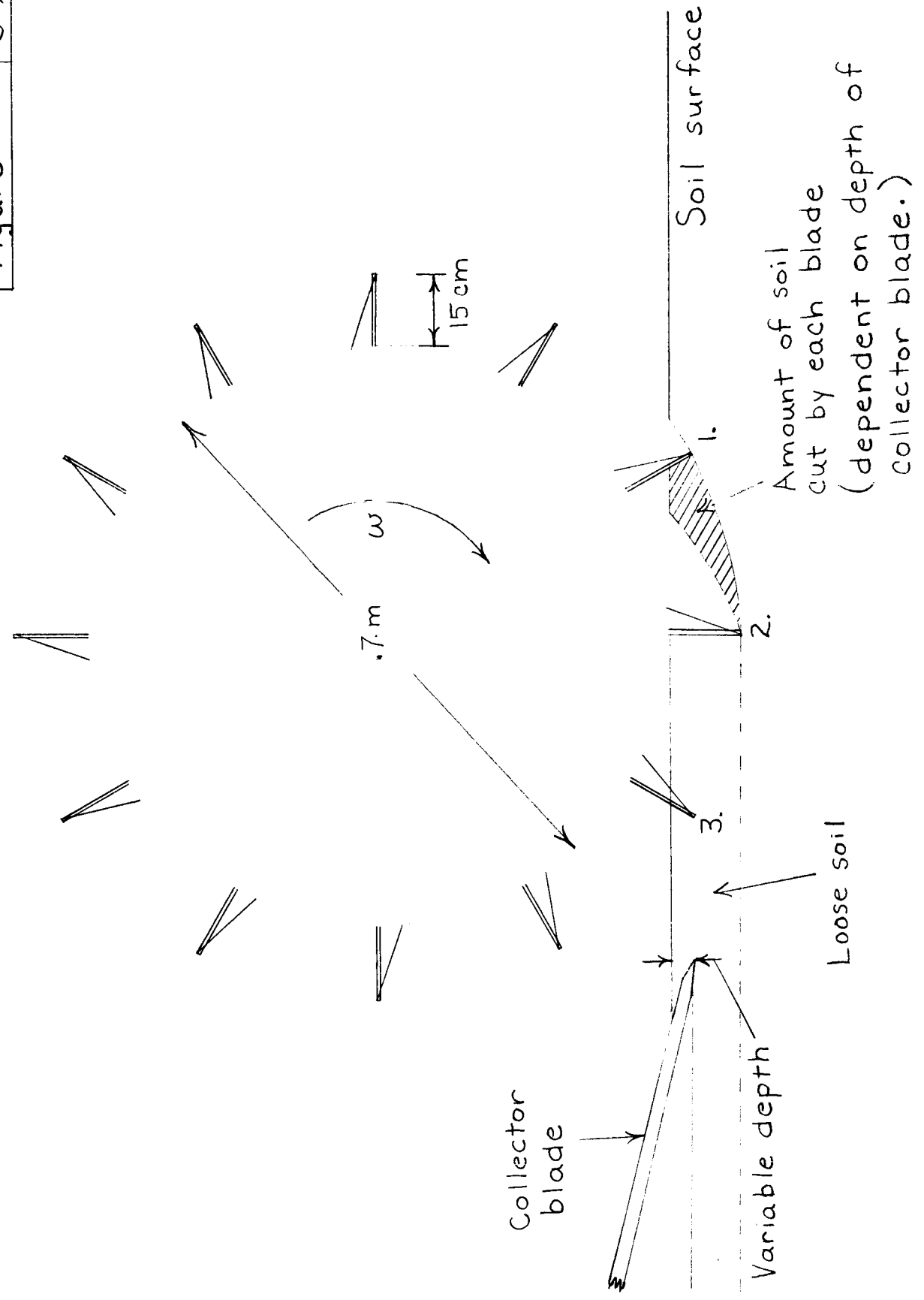


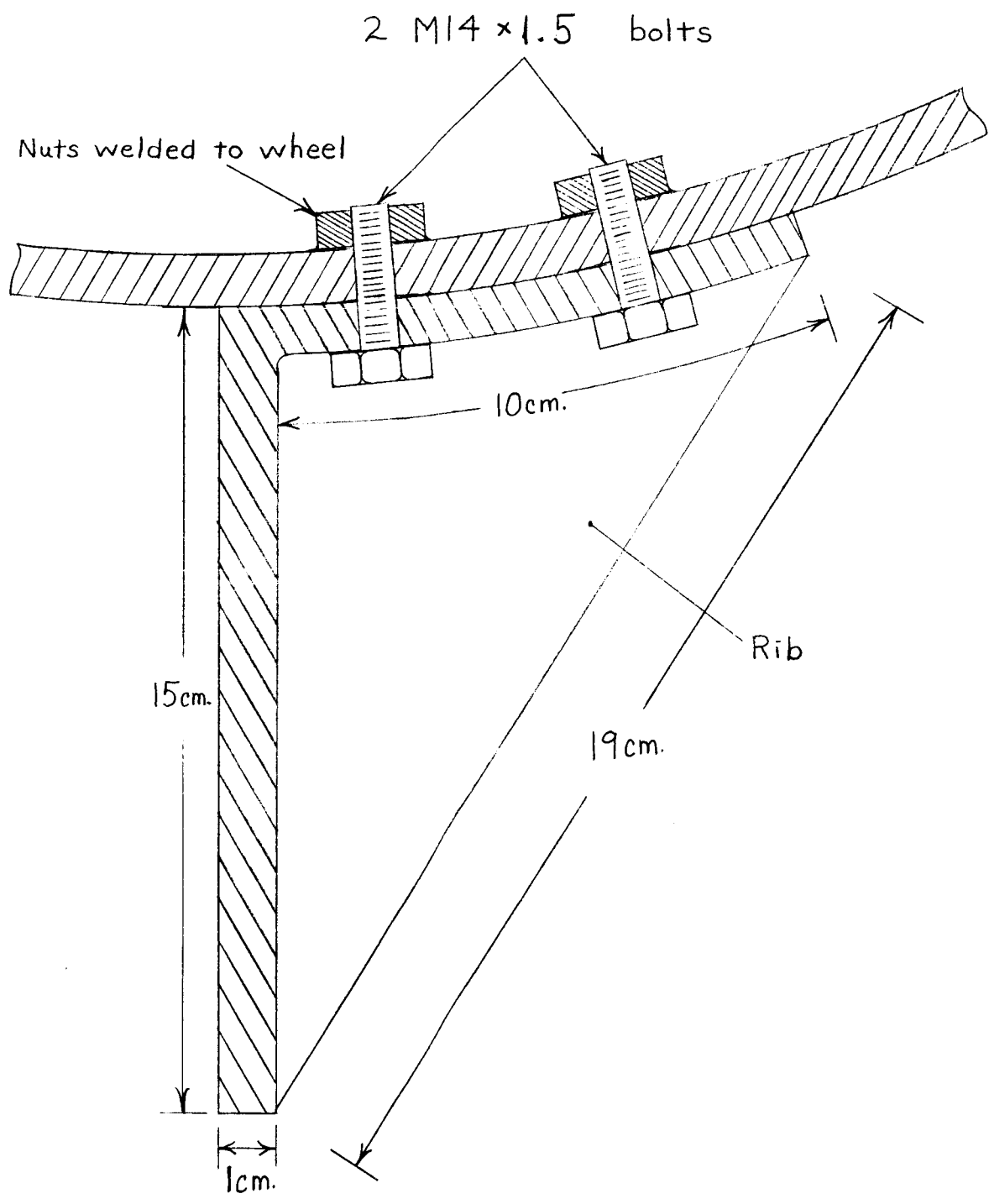
Collection Blade

Drawn by DPC

No. 12 Set 1:25

Side of Front Wheel		
Russ Anderton	#13a	
Figure	5-29-86	





Blade attachment	
Russ Anderton	#13b
Full scale	5-29-86

GEAR DRIVE

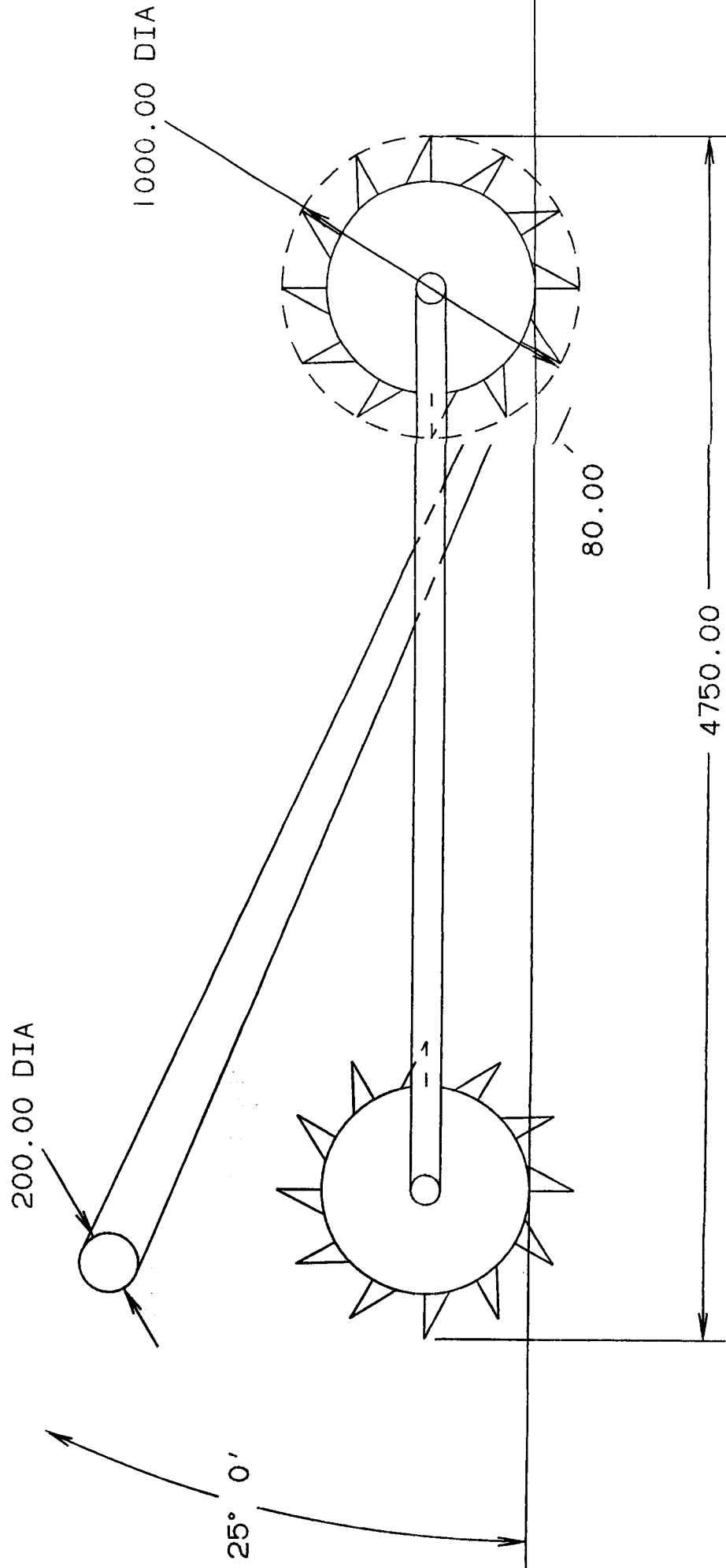
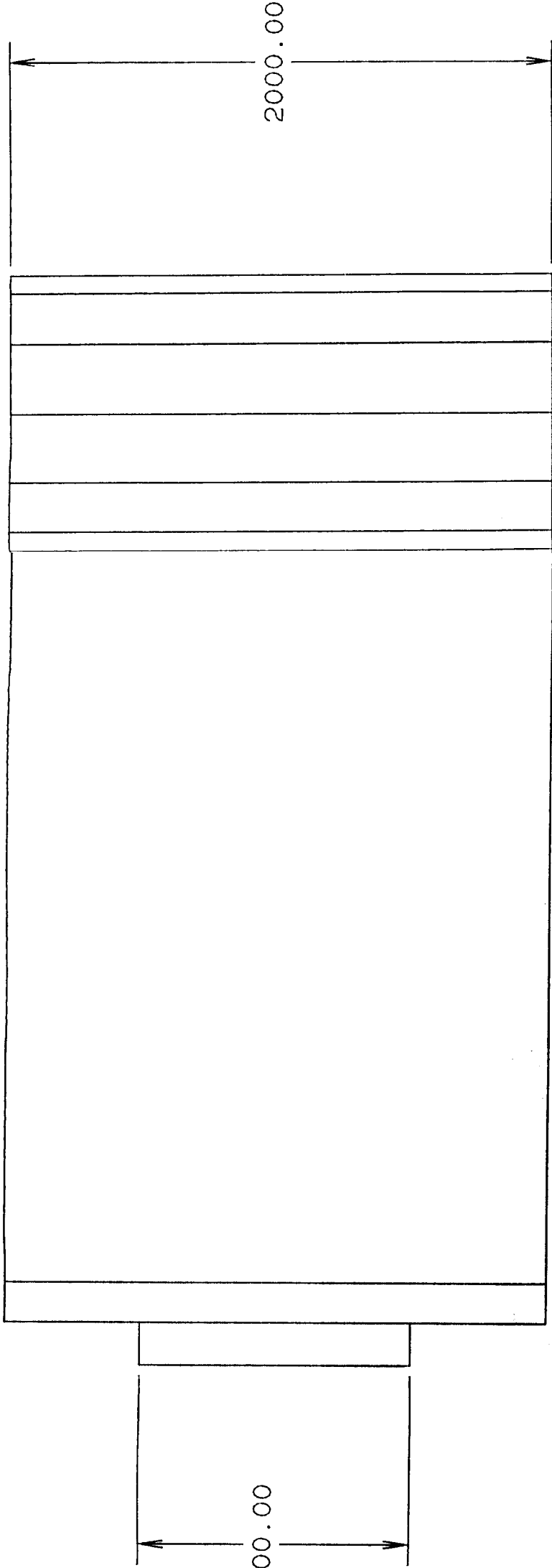
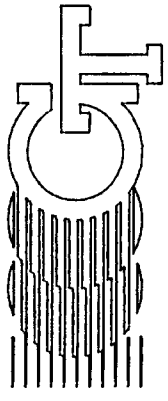
MOUNTING
BRACKET
4-PLACES

WORM GEAR

DRIVE MOTOR

BLANKING

TITLE: STEERING MECHANISM	
DRAWN BY: PAUL ALLEN	NO. 14
SCALE: NONE	DATE: 01 JUN 86



D/E MM

GEORGIA TECH

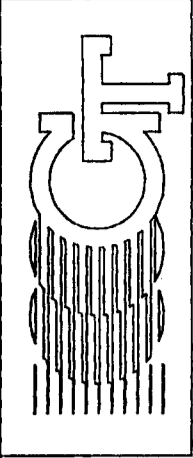
COLLEGE OF ENGINEERING

TITLE: MINER ASSEMBLY

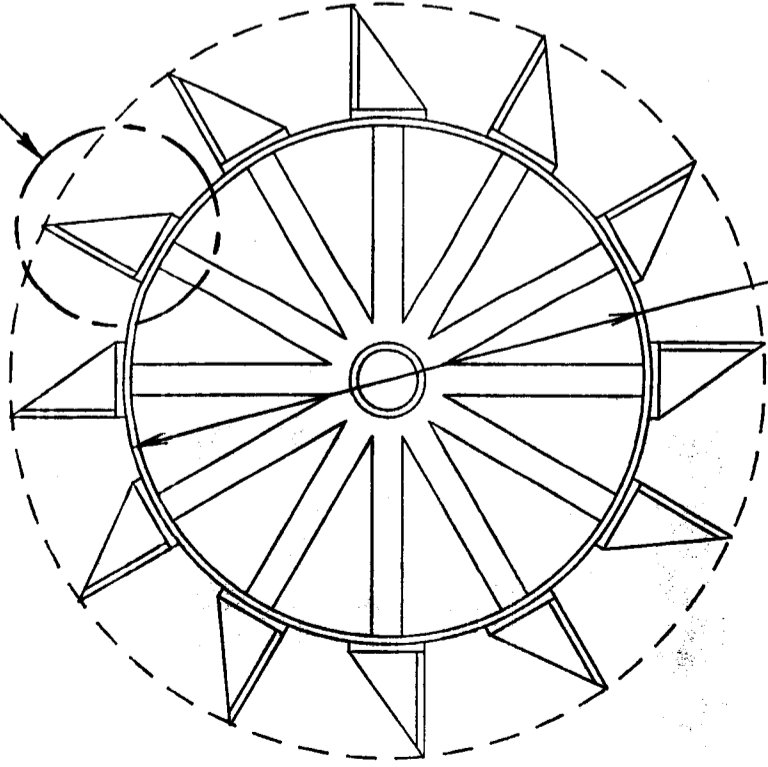
DESIGN: DATE

CHECK: DATE

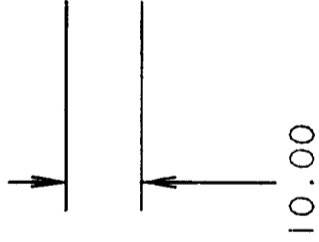
DRWG NO. 15 SCL



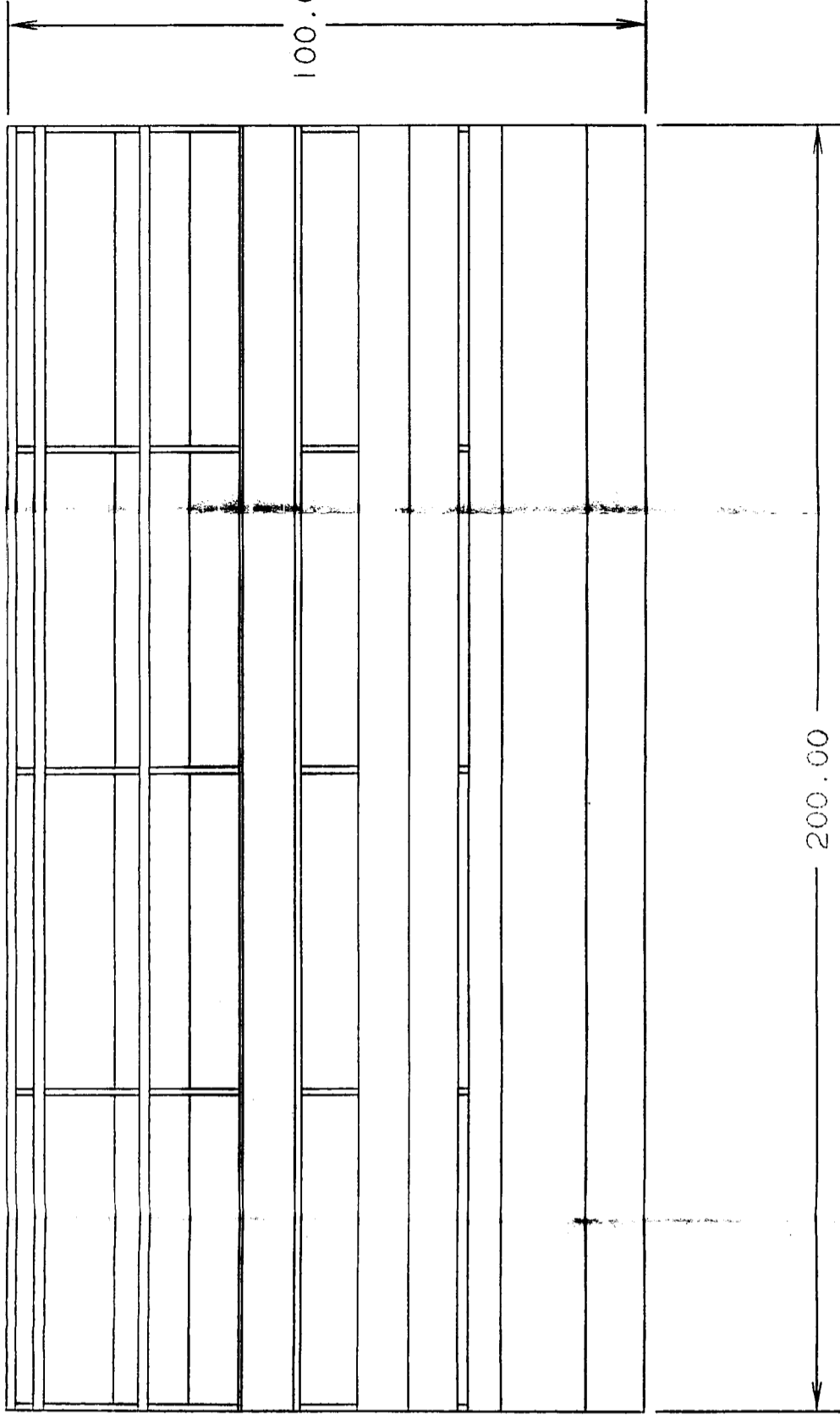
DETAIL A (SEE DRWG NO. 17)



70.00 DIA



10.00

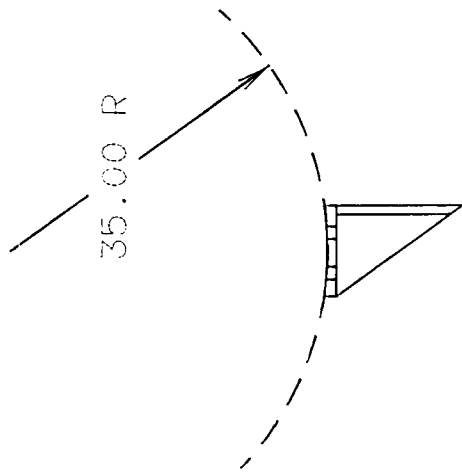
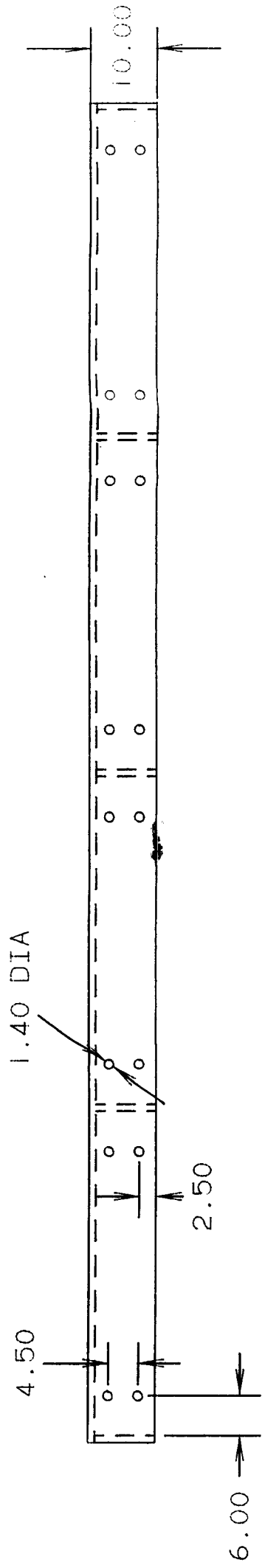
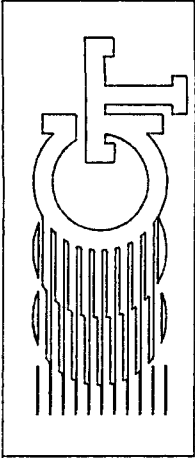


200.00

100.00

ALL DIMENSIONS IN CM.

GEORGIA TECH
COLLEGE OF ENGINEERING
TITLE: FRONT WHEEL
DESIGN: DPC
CHECK:
DATE 31MY86
DATE
DRWG NO. 16
SCL 1.0



DETAIL A

ALL DIMENSIONS ARE CM

GEORGIA TECH			
COLLEGE OF ENGINEERING			
TITLE: WHEEL BLADE 1 OF 12			
DESIGN: DPC	DATE	29MY86	
CHECK:	DATE		
DRWG NO. 17	SOL	1.2	

9. ACKNOWLEDGMENTS.

We wish to thank the following for their help with this project:

Jerry Edwards - U.S. Forestry Department; Atlanta, Georgia

Barney Roberts - NASA; Houston, Texas

Dr. George Sowers, Ph.D. - Civil Engineering, Georgia Institute of Technology; Atlanta, Georgia

Dr. Neal D. Williams, Ph.D. - Civil Engineering, Georgia Institute of Technology; Atlanta, Georgia

Dr. Leland Riggs, Ph.D. - Civil Engineering, Georgia Institute of Technology; Atlanta, Georgia

Dr. Robert C. Bachus, Ph.D. - Civil Engineering, Georgia Institute of Technology; Atlanta, Georgia

C.J. Swafford - Mechanical Technician, Georgia Tech Research Institute; Atlanta, Georgia

Robert M. Flipppo - Mechanical Research Corporation; Pasadena, California

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Apollo 11 Preliminary Science Report Washington, D.C.: National Aeronautics and Space Administration-Scientific and Technical Information Division; 1969.

Progress Report - April 10, 1986

Soil Extraction System for Lunar Mining Vehicle

Our ME 4182 group met for the first time April 3 and chose project number four dealing with the excavation of soil from the moon. Our first group meeting was Monday, April 7. During the one and one-half hour session, we discussed the project's goals and priorities as we saw them. This project is an extension of a project begun by a past ME 4182 group. Our priorities lie with the actual digging or excavation mechanism and the means of travel for the vehicle.

The past ME 4182 group suggested the vehicle's digging mechanism would move the vehicle along the terrain as it dug. Our group decided to design a vehicle that would travel regardless of whether or not the digging mechanism was in use. Such a design would allow for travelling fully loaded to a desired location without having to dig up soil at the same time. Also, unless the method of travel is separate from the digging mechanism, soil would be excavated everywhere the vehicle travelled.

Several methods were discussed dealing with the actual digging mechanism. Ideas included a drill or a ditch-digger type mechanism that would drop the soil onto a conveyor system. These and other ideas will be discussed in more detail at later meetings.

Finally, after much deliberation, the group decided upon a title for the project. The deliberation dealt with creating a title that would accurately describe our project. The title decided upon was "Soil Extraction System for Lunar Mining Vehicle". Our next group meeting is Monday, April 4 at 6:00 pm.

17 April, 1986
Group 4
Soil Engagement

PROGRESS REPORT

Group 4 was divided into 3 subgroups for the initial purpose of researching the background needed for design development. One group will investigate the present day terrestrial excavation equipment, the second will look into soil mechanics, and the third will research the lunar environment and past mechanization/materials used on the moon.

TERRESTRIAL EXCAVATION - The group contacted numerous Atlanta construction firms and should be receiving brochures and information within a week. The GT library was searched for information, but only outdated material was found. Future plans are to contact the Civil Engineering Department to locate any professors doing work on new excavation equipment.

SOIL MECHANICS - Through a recent film and several books, initial information has been gathered on the nature of the lunar soil. Information has been attained on the composition, granularity, density, and layered nature of the lunar soil. Time, shock waves, and hypervelocity impacts are factors responsible for the existence and nature of the soil. NASA equations modeling the lunar soil have been obtained. Further pursuits are directed towards learning about how soil is modeled for designing terrestrial excavation equipment.

LUNAR ENVIRONMENT - Some preliminary designs were proposed for later acceptance or dismissal pending developments from research. Information was found on soil penetrability and other aspects of the lunar surface that will directly affect motion of digging apparatus. and

Progress Report, Group 4

April 24, 1986

Work was continued in three separate groups on ideas brought up in the weekly conference and in brainstorming sessions.

One group continued to research modern earth excavation machinery. Brochures were obtained from local vendors showing equipment manufactured by Caterpillar and John Deere. The VSMF was also checked in the library for examples of excavation equipment. One group member contacted the tillage lab at Auburn. Several books, such as the Excavation Handbook, have been checked out from the library.

The soil mechanics research continued with further study of related books. Also, the VSMF was used for searching agricultural products and industry standards. Documents from the NASA notebook were not in the library stock. This week's tasks include learning how to apply lunar penetrability data to practice engineering problems. Contact will be made with the C.E. department to aid in this study.

The group studying the lunar surface created several more design alternatives. Possible venues for further research now include the Department of the Interior. Anchoring in the lunar soil seems to be a possibility for traction and guidance. In a conference with Dr. Bachus of the C.E. department, the meaning of lunar soil parameters was outlined for use in force diagrams involving anchoring and traction. A stress diagram was developed. The groups will meet again April 28 to discuss new information.

1 MAY, 1986
Group 4
Soil Engagement

PROGRESS REPORT

The entire group went to see Jerry Edwards in the engineering department of the USDA Forestry Service to discuss equipment utilized by his department. The use of a plow with a certain application as moving anchor behind or underneath a continuous miner. We also sounded out his opinion on using augurs or some other kind of anchor, which he agreed would be good for mostly stationary types of digging, such as covering a lunar shelter. In the next week the group will see Jimmy Whitfield about detailed use of plows and their manufacture. The preliminary outline was prepared this week.

CONSTRUCTION EQUIPMENT - This subgroup visited the USDA Forestry Station to get pictures of the fire-line plow, and discussed and sketched additional proposed designs.

LUNAR ENVIRONMENT - This group arranged the consultations with Jerry Edwards and Jimmy Whitfield, and work was done on figures for working with lunar soil. Preliminary production figures (material/year) were established.

SOIL MECHANICS - Efforts are now concentrated on application of the information we have gained about lunar soil characteristics. A visit with C.E. professor, Dr. Sawers, led to clarification of how to use his equations and modify them for lunar soil, which is cohesionless. He also directed us to some very promising C.E. journal articles by Ronald Scott. We also discussed the fire plow in cohesionless soil with Jerry Edwards. Calls were made to Paul Corcoran (Caterpillar Engineer) and Dean Freitag (LRV Design), both of which led to further resources and information which will be studied this week.

Progress Report

Group 4

May 8, 1986

The group was split into two sections this week. One group worked on heavy digging in a localized area. The other group studied developing a continuous strip mining system for the collection of soil.

The heavy digging group discussed the idea of augers for anchors, but discounted the idea as too power consuming and not reliable in all soils. Two new ideas were explored. One idea used counterbalances on digging arms, and the other used draught blades on the opposite end of the vehicle from the scoop. If the digging action caused the vehicle to pull forward, the draft blade would dig in to provide traction. Several applications of these blades as anchors and scoops were outlined.

The mining group began to evaluate ideas that had been generated over the past weeks. Several ideas utilized blades on wheels or cylinders. The problem of where (on the circumference of the wheel) the soil would release for collection was studied. This involved calculations of centrifugal acceleration and gravitational forces. Also, two promising documents on the mechanics of soil cutting blades were found. These should aid in calculation of draught and vertical blade forces. These will be used to determine torque requirements for different cutting wheels.

Progress Report

Group 4

May 15, 1986

The heavy digging and strip mining groups continued their work in their respective areas this week. The heavy digging group narrowed its designs down to two proposals. Two groups within each group were designated, each to work on one design. A number of parameters will be analyzed by the end of this week. They include: approximate weight of bucket, draft force, break-out force, rock capabilities, complexity, reliability, versatility, cycle time, mobility, and vehicle considerations. After these parameters are found, one of the two proposed designs will be chosen, and it will be analyzed further. A visit was made with Dr. Riggs of the civil engineering department. He helped members of the group to better understand the digging forces applied by a backhoe bucket.

The mining group also visited the civil engineering department this week. Dr. Williams answered many questions they had. The mining group spent most of its time this week building models in a machine shop that would help them test their various design ideas. These designs will hopefully lead to a final decision on a surface mining design.

The groups hope to decide on specific designs next week and begin their final design work.

Progress Report

May 22, 1986

Group 4

Group 4 began to put together the initial pieces of the final report this week. Three of the group members were experienced or quickly became experienced at using a CAD system. They spent most of the week drawing the designs that have been decided upon by the digging and mining groups. Other members were busy working on calculations necessary to complete these designs. Force calculations on the interaction between the soil and cutting blades is an important area still being studied. Results of this study will enable group members to make decisions of the power requirements for the machinery.

While some members are working on drawings and calculations, other members of Group 4 began working on a rough draft for the final report. Not all areas of the report are ready to be finalized; however, topics such as the introduction, information from past research, and theoretical calculations can be written. Next week the group will assemble the information from its members to form an initial rough draft of the project report.

Group 4

Progress Report

May 29, 1986

This past week Group 4 tried to put on paper the results of this past quarter's design project. Much work has been done, and as a result, much was to be written. Most of the background material had already been written, but descriptions of final design decisions as well as discussions of the methods used to reach these decisions were missing. While much of the group worked to finish these tasks, other members continued working on drawings displaying various aspects of the design. The school's CAD system is still giving the group problems because of troubles with the printer device. Some of the drawings are now being prepared on a drafting board.

The group has learned the past few days that the major problem now faced is not writing the report, but putting the various aspects of the final report together in an orderly manner. The members have attempted to write their parts of the report with the same word processing program, PC Write. This will facilitate combining different parts of the final report. After much work, a rough draft of the final report is ready. What remains to be accomplished are final touches to make the report neat and easy to follow.

A Record of Invention is a very important legal document, and proper care in its early and complete preparation will save much time and inconvenience in the future should the invention ever become involved in a controversy. Separate Record of Invention forms must be prepared for each distinct invention. This Record of Invention form was prepared to fit all possible conditions and therefore contains questions which may not be applicable to certain inventors. If a question cannot be answered, mark it "does not apply," "no" or "none," etc.

1. Give name in full - JOHN ALLEN DOE or JOHN A. DOE, and position. J. ALLEN DOE will not be accepted by the Patent Office.
2. Give complete details - name, address, phone and extension, city, state.
3. Indicate date first employed.
4. Give your present address. This is your residence address.
5. If No. 4 is a temporary address, give a permanent address where mail can be sent and forwarded regardless of your temporary location. If No. 4 is your permanent address, No. 5 may be marked "Same as No. 4."
6. The question as to whether an invention was actually conceived by one person or by the cooperation of several persons is vitally important to the validity of the patent. Incorrect decisions of sole or joint inventorship will result in an invalid patent. Personal feelings and friendships must not influence the answer to this question. To merely present a problem does not make one an inventor. He must contribute to the solution of the problem to be classed as an inventor. A mechanic who builds a device under the supervision of an inventor is not a joint inventor. If there is any doubt, obtain assistance.
7. A short title briefly describing the invention. For instance, "Method and Apparatus for Preparing Segmental Carbon Rings," "Uni-directional Antenna," or "Packing Gland Wrench."
8. A list is desired of all pertinent material which forms the "Disclosure of Invention," thereby relating the Record of Invention to the disclosure. Give accurate print reference numbers, photo numbers, etc., identifying sketches by date and/or some other reference.
9. This is the date on which the general "idea" was "born". State briefly circumstances relative to conception. For example, on 2 February 1945, I saw accident to John A. Doe while working on circular saw and need for a guard was apparent. Immediately made sketch for a suitable guard and presented it to James Smith, my foreman. Or, test for signal generator was unsuccessful for several months. On 21 January 1946 made change in circuit by changing capacity and location of by-pass condenser and installing suitable chokes.
10. This date may coincide with No. 9, will never be earlier, but may be later.
11. This date also may be the same as No. 9 and No. 10, or different from either one, but never earlier than No. 9.
12. The disclosure of the invention may be by orally telling someone of your invention, by showing and explaining descriptions and drawings, or by demonstrating the operation of a model or full scale device. The person(s) to whom an invention is disclosed may make an excellent witness if the invention is ever involved in a controversy and it is to the inventor's advantage to see that his invention is understood. It is preferable to ask the witness to read and sign the description and drawings, using the statement "Disclosed to and understood by my this _____ day of _____ 19____." Notebook entries should be handled in a like manner. Indicate persons' position, connection with outside commercial firm, or other pertinent information. State if orally, by sketch or other form.
13. If no model or full scale device was made, so state. Answers to Nos. 14 to 18 inclusive should be marked "does not apply."
14. Indicate where model or full scale device may be seen in the event the attorney desires to witness its operation.

15. State whether first tests were successful or unsuccessful and briefly outline results.
16. Witnesses to tests are important. Identify them so they can be located later if necessary.
17. Tell briefly results of later tests, and give data on witnesses.
18. It is desirable to record carefully the results of all tests and to preserve such records. A copy of these test results may form a part of the invention disclosure to assist the attorney in preparing the case.
19. In related inventions, much repetition can be avoided by reference to the previous invention(s), if any. Identify such records or reports by title, inventor, date, number, etc.
20. Most inventions are improvements over existing devices. References to the devices which have been improved are very helpful to the attorney preparing the patent application. Such references may consist of patent numbers, issued patents, pending applications, articles published in technical magazines, listing in catalogs or trade journals, or any other published record of the invention.
21. The law provides that after an invention has been in "Public Use" for more than one year it becomes public property and cannot be protected by Letters Patent. The question of public use is a debatable one. The same is true of experimental use, and it is sometimes difficult to ascertain when experimental use ceases and public use begins. A statement as to the use of your invention with sufficient details as to the circumstances will assist in determining whether public use exists. Differentiate, if possible, between use by the Government and commercial use.
22. Where details of the invention have been released to a firm or activity, it is important to record the data to clearly fix the facts of the release of information. The firm may be under contract with a federal agency. If so, the contract number should be given.
23. If classified, the proper classification is to be stamped on both the top and bottom of each page.
- V. Primm*

195 E. Place

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R.D. 1 Valatie

N.Y. 12184

This is an important legal document. Read instructions carefully before filling in data.

PROJECT NO. _____ RECOMMENDED SECURITY CLASSIFICATION _____ REC. OF INV. NO. _____

1. NAME OF INVENTOR Bernhard W. Kuchel POSITION Student

2. DEPARTMENT OR DIVISION School of Mechanical Engineering, Georgia Institute of Technology, Atlanta, GA

3. DATES OF EMPLOYMENT 8/82 (Undergraduate Student)

4. PRESENT ADDRESS (No, Street, City, County, State) Ga. Tech. Box 37496 Atlanta, Ga. 30332 TELEPHONE (404) 676-0647 PERMANENT OR UNTIL 6-14-86

5. PERMANENT ADDRESS (No, Street, City, County, State) 103 Ellen Way Brandon, FL 33511 TELEPHONE (813) 689-2553

6. NAMES (S) AND ADDRESS (ES) OF CO-INVENTORS (If any)
James F. Clark 209 Hardeman Ave. Ft. Valley, Ga / Don P. Clark 1226 Roslyn Ave, Bradenton, FL / Jeffrey D. Keesee 1180 Old Village Run Dunwoody, Ga / George E. Aldridge, Jr. 6333 Rockland Rd. Lithonia, Ga / Paul B. Allen 2119 Gunstock Dr. Stone Mountain, GA

7. DESCRIPTIVE TITLE OF INVENTION Method and apparatus for excavating and mining the lunar surface.

8. LIST DRAWINGS, SKETCHES, PHOTOS, REPORTS, DESCRIPTIONS, NOTEBOOK ENTRIES, ETC., WHICH SHOW OR DESCRIBE INVENTION
Attached report.

9. EARLIEST DATE AND PLACE INVENTION WAS CONCEIVED (Brief outline of circumstances)
April 4, 1986 realized need for lunar soil manipulation system to support NASA lunar base phase.

10. DATE AND PLACE OF FIRST SKETCH, DRAWING OR PHOTO
April 11, 1986

11. DATE AND PLACE OF FIRST WRITTEN DESCRIPTION
May 10, 1986

12. DISCLOSURE OF INVENTION TO OTHERS			
NAME, TITLE AND ADDRESS	FORM OF DISCLOSURE	DATE AND PLACE OF DISCLOSURE	WAS SIGNATURE OBTAINED (YES OR NO)
<u>Mr. J. Eversell, Lecturer Ga. Tech.</u>	<u>verbal sketch</u>	<u>4/4/86 Ga. Tech.</u>	<u>No</u>

12.A IMPORTANT - HAVE ANY PUBLICATIONS OR REPORTS BEEN MADE ON THIS INVENTION?
Only present report at 6/7/86.

13. DATE AND PLACE OF COMPLETION OF FIRST OPERATING MODEL OR FULL SIZE DEVICE
5/15/86 Ga. Tech. - model

14. PRESENT LOCATION OF MODEL
Ga. Tech.

15. DATE, PLACE, DESCRIPTION AND RESULTS OF FIRST TEST OR OPERATION
5/20/86; Ga. Tech. Solar Research Facility, Lunar Mining Device model seemed successful in operation with conventional earth soils.

16. NAMES AND ADDRESSES OF WITNESSES OF FIRST TEST

Paul E. Allen 2119 Gunstock Dr., Stone Mountain, GA 30087
 Russel M. Anderson 4102 Kingstrap Rd., Stone Mountain, GA 30083
 George E. Aldridge, Jr. 6333 Rockland Rd., Lithonia, GA

17. DATE, PLACE, DESCRIPTION AND RESULTS OF LATER TESTS (name witnesses)

None

18. IDENTIFY RECORDS OF TESTS AND GIVE PRESENT LOCATION OF RECORDS

Video Tape of tests in Paul E. Allen's possession

19. PRIOR REPORTS OR RECORDS OF INVENTION TO WHICH INVENTION IS RELATED

None

20. OTHER KNOWN CLOSELY RELATED PATENTS, PATENT APPLICATIONS AND PUBLICATIONS

PATENT OR APPLICATION NO.	DATE	TITLE OF INVENTION OR PUBLISHED ARTICLE	NAME OF PUBLICATION
—	—	Tree Spade Device	Vermeer Manuf. Co
/			

21. EXTENT OF USE: PAST, PRESENT AND CONTEMPLATED (Give dates, places and other pertinent details)

To excavate and mine on lunar surface by NASA.

22. DETAILS OF INVENTION HAVE BEEN RELEASED TO THE FOLLOWING COMPANIES OR ACTIVITIES

NAME AND ADDRESS	INDIVIDUAL OR REPRESENTATIVE	CONTRACT NO.	DATE
None			

SIGNATURE OF INVENTOR

Benhard V. Kuehl

DATE

6/1/86

(Attach to Record of Invention Part I)

REC. OF
INV. NO. _____

This Disclosure of Invention should be written up in the inventor's own words and generally should follow the outline given below. Sketches, prints, photos and other illustrations as well as reports of any nature in which the invention is referred to, if available, should form a part of this disclosure and reference can be made thereto in the description of construction and operation.

1. INVENTOR NAME (S)

Bernhard W. Kuchel

2. TITLE OF INVENTION

Lunar soil excavation and mining system.

For answers to following questions use remainder of sheet and attach extra sheets if necessary.

3. GENERAL PURPOSE OF INVENTION. STATE IN GENERAL TERMS THE OBJECTS OF THE INVENTION.

4. DESCRIBE OLD METHOD (S) IF ANY, OF PERFORMING THE FUNCTION OF THE INVENTION.

5. INDICATE THE DISADVANTAGES OF THE OLD MEANS OR DEVICE (S).

6. DESCRIBE THE CONSTRUCTION OF YOUR INVENTION, SHOWING THE CHANGES, ADDITIONS AND IMPROVEMENTS OVER THE OLD MEANS OR DEVICES

7. GIVE DETAILS OF THE OPERATION IF NOT ALREADY DESCRIBED UNDER 6.

8. STATE THE ADVANTAGES OF YOUR INVENTION OVER WHAT HAS BEEN DONE BEFORE.

9. INDICATE ANY ALTERNATE METHODS OF CONSTRUCTION.

10. IF A JOINT INVENTION, INDICATE WHAT CONTRIBUTION WAS MADE BY EACH INVENTOR.

11. FEATURES WHICH ARE BELIEVED TO BE NEW.

12. AFTER THE DISCLOSURE IS PREPARED, IT SHOULD BE SIGNED BY THE INVENTOR(S), AND THEN READ AND SIGNED AT THE BOTTOM OF EACH PAGE BY TWO WITNESSES USING THE FOLLOWING STATEMENT:

"DISCLOSED TO AND UNDERSTOOD BY ME THIS _____ DAY OF _____ 19____
SIGNATURE _____"

Note 1: CO-INVENTORS (cont'd) Coy McCoy 1821 Cedarwood Rd. Birmingham Ala. / Russel M. Anderton 4102 Kingstrapp Rd. Stone Mountain, GA / Douglas E. Peace R.D. 1, Valatie, N.Y. 12184

See attached report for complete description.

REC. OF
INV. NO. _____

Bernard V. Kichel
INVENTOR
June 2 19 86

DISCLOSED TO AND UNDERSTOOD BY ME
ON THIS _____ DAY OF _____ 19 _____

WITNESS

DISCLOSED TO AND UNDERSTOOD BY ME
ON THIS _____ DAY OF _____ 19 _____

WITNESS